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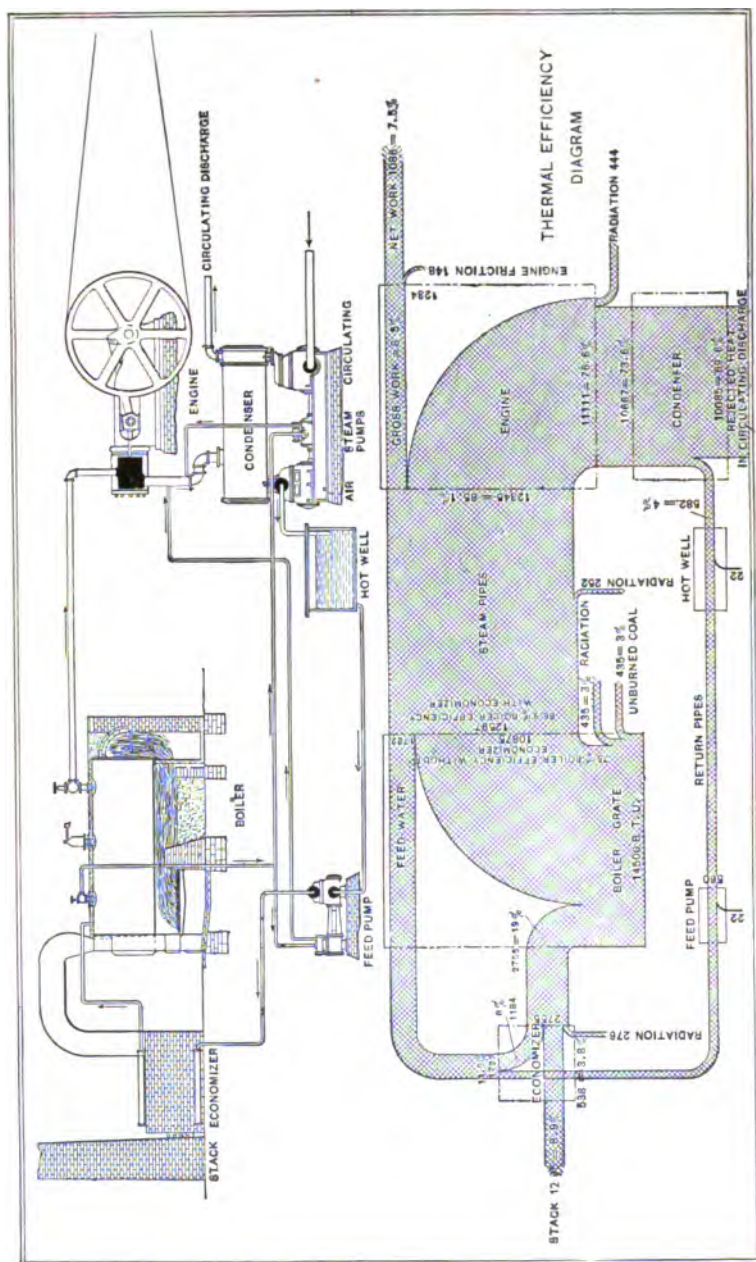
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STEAM-BOILERS
THEIR THEORY AND DESIGN



Thermal-efficiency Diagram.

STEAM-BOILERS

THEIR THEORY AND DESIGN

BY

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THIRD EDITION

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THIS WORK
IS DEDICATED TO
MY WIFE.

PREFACE

IN presenting this book to the Engineering Profession and to fellow students in practical science, the Author desires to state that no claim is made for originality. In fact it would be nearly impossible, if not quite so, to write a work on this subject which could be considered original.

These pages comprise, in book form, a series of lectures delivered to the Senior Class of the Rensselaer Polytechnic Institute, Troy, New York, rewritten and divided into chapters. The only originality claimed for the work is the effort to cover such points as in practical office work may be found to be perplexing.

No one should attempt to design a steam-boiler until he has had some experience in, or personal acquaintance with, boiler-shop practice. There are many things in the actual putting together of the parts of a boiler which cannot be clearly described, and for just such things even a short shop experience would be most valuable.

The Author acknowledges obligations for the free use which he has made of literature on the subject; and, while many references are mentioned by name, he now expresses his thanks to those to whom special reference has not been made.

H. DE B. PARSONS.

NOTE ON THE SECOND EDITION

IN preparing this book for the second edition, the Author has made some changes to which he desires to call special attention.

The frontispiece which has been added illustrates the dissipation of the heat produced by the combustion of a fuel on a grate. The upper part of the cut shows a boiler, engine, condenser, hot-well, feed-pump, and economizer, all connected by piping. The lower part shows the heat flowing in streams, drawn to scale, under an assumed set of conditions. Where the heat is rejected from such a cycle, so as not to be recoverable, the streams appear as if running off into space. In the cut, the heat at the grate corresponds to the total heat of combustion of one pound of an assumed coal. By a combination of an engine and a boiler, it is not possible to transform all of this heat into useful work, as is proven by the study of thermodynamics. From any boiler or engine trial, when sufficient data have been obtained, a similar diagram can be constructed to illustrate the conditions so found.

Fig. 8 illustrates one of the latest arrangements of a locomotive fire-box for burning liquid fuel. The arrangement is designed to provide a thorough mixture of the oil and air, and allow time for the combustion to take place before the products are drawn into the small tubes and chilled below the temperature of ignition.

In the text, under Liquid Fuels, some changes have been made and attention called to the valuable report of the Liquid Fuel Board of the United States Navy Department, 1904.

Fig. 20 illustrates one of the most approved English methods for the brick setting of a Lancashire boiler.

The general plan of the book remains unchanged.

H. DE B. PARSONS.

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STEAM-BOILERS

CHAPTER I

PHYSICAL PROPERTIES

Solid Bodies. Fluid Bodies. Liquid Bodies. Gaseous Bodies. Perfect Gas. Laws of Gases. Heat. Conduction. Convection. Radiation. Mechanical Equivalent of Heat. Absolute Zero. Specific Heat. Latent Heat. Total Heat of Evaporation. Weight of Water. Boiling. Relative and Specific Volumes of Steam. Factor of Evaporation.

THERE are two principal states in which all bodies are found, namely, "Solids" and "Fluids." Fluids may again be divided into "Liquids" and "Gases."

Solid Bodies may be defined as those which will resist a longitudinal pressure, no matter how small that pressure may be, without being supported by a lateral pressure.

Fluid Bodies may be defined as those which will not resist such a longitudinal pressure.

Liquid Bodies may be defined as those which will only partly fill a closed vessel, while the rest of the vessel may be either empty or contain some other fluid.

Gaseous Bodies may be defined as those which will expand and completely fill a closed vessel, no matter how small a portion may be introduced. Gases are thus distinguished by their power of indefinite expansion.

A Perfect Gas may be described as one which obeys exactly the laws of Mariotte and of Gay-Lussac. Such a perfect gas is now known to be ideal, and the so-called permanent gases only approximate in their action to these laws in accordance with their degree of perfection.

Laws of Gases. First Law (Mariotte or Boyle): "At constant temperature, the volume of a portion of gas varies inversely as the pressure." That is, $pv = \text{constant}$.

Second Law (Gay-Lussac, Charles, or Dalton): "At constant pressure, the volume of a portion of gas varies directly as the absolute temperature." That is, $v = \text{constant} \times \tau$.

Heat. There are many words, such as "hot," "warm," "tepid," "cool," "cold," which are used to denote different sensations that indicate a corresponding condition of the object with respect to the heat which it is said to contain. These conditions or series of states are called "Temperatures," and from the facts as found in nature it must be admitted that there exists an infinite number of these intermediate states or temperatures.

The temperature of a body, therefore, indicates how hot the substance is. These temperatures are accompanied in each body by certain conditions as to the relations between density and elasticity. In general, the hotter the body, the less is its elasticity of figure and the greater is its elasticity of volume.

Heat may be considered as a "mode of motion," and is generally recognized to be a vibratory motion of the particles composing any body.

Heat is transferable from one body to another, that is, one body can heat another by becoming less hot itself. This transfer of heat between two bodies tends to bring them to a state called "uniform" or "equal" temperature. At uniform temperature this transfer of heat ceases.

Heat is transferred from a warmer body to a colder body by one of three processes, namely, "Conduction," "Convection" and "Radiation."

Conduction is the transference of heat between two contiguous portions of matter at different temperatures. Convection is the distribution of heat by a movement of a portion of a fluid within its own mass. Such a movement is called a convection current. Radiation is the transference of heat from one body to another at a distance, through an intervening transparent medium.

Heat is one of the forms of energy, since it may be transformed into mechanical work.

As the condition of heat is a condition of energy, and is capable of effecting changes, it may be indirectly measured, so as to be

expressed as a quantity by means of one or more of the directly measurable effects which it produces.

When the condition of heat is thus expressed as a quantity, it is subject, like all other forms of energy, to a law of conservation.

Since the properties of all substances vary with their temperatures, it has become customary to make use of two of these variations to indicate particular temperatures as points of reference. These two variations were selected because they were abrupt and well defined, and are:

First. The temperature at which ice melts under one atmosphere of pressure, equivalent to 14.7 pounds per square inch or a barometric height of 29.95 inches. As this temperature varies but slightly with changes of pressure, it can be easily reproduced under ordinary conditions.

Second. The temperature of steam generated from water when boiled under one atmosphere of pressure.

Occasional use is made of other changes of state, which take place at temperatures more or less well defined, such as the melting-points of certain metals and alloys.

Ordinary temperatures are recorded from the reading of a mercurial thermometer.* For higher temperatures use is made of the air-thermometer or some form of pyrometer.

Heat and work are mutually convertible in a fixed ratio, known as the "Mechanical Equivalent of Heat." The relationship existing between heat and work, was demonstrated by various experiments, the most noted being those of Benjamin Thompson, better known as Count Rumford (1753-1814), Sir Humphry Davy (1778-1829), Sadi Carnot (1796-1832) and Henry A. Rowland (1848-1901). In 1842, Dr. Mayer, of Heilbronn, is said to have first introduced the expression "Mechanical Equivalent of Heat," and in the year following Dr. Joule, of Manchester, measured this equivalent. The value placed by Joule was 772 foot-pounds of work as equivalent to one British thermal unit. This result is still in use,

* In order accurately to read a mercurial thermometer, when the scale is not on the same plane with the column of mercury, it will be found convenient to hold a small looking-glass behind the column of mercury; then, when the eye is so reflected that the centre of the pupil is coincident with the top of the mercury, the eye will be at right angles to the mercury and also to the scale.

although later experiments show that 778 foot-pounds is nearer the true figure. This latter result will be used in this work except where otherwise stated.

The British thermal unit is the quantity of heat required to raise one pound of pure water at its maximum density one degree Fahrenheit. The temperature of water at maximum density is very nearly 39.2°F .

The Absolute Zero. In order to simplify calculations with respect to the action of perfect gases, all the formulæ are based on a scale of absolute temperatures. These absolute temperatures express the heat of a body on a scale beginning at a point known as the "absolute zero."

The absolute zero is a theoretical point on this temperature scale that is fixed by assuming that the law of gases as determined by experiment remains constant throughout the whole range of temperatures.

It may be said to be the temperature point corresponding to the disappearance of gaseous elasticity, or the point at which the expression p_v for a perfect gas becomes zero.

When a portion of dry air is heated from the freezing-point of water (32°F .) to the boiling-point (212°F .), that is, its temperature has been raised through 180°F ., it will expand to 1.365 of its original volume. Therefore, when the gas has been heated through 493.2° , it will expand to twice its original volume. If, therefore, the same law holds true for cooling, and the temperature be lowered 493.2° from the freezing-point (32°), the volume of the gas will be reduced to zero. But the law of expansion and contraction of the so-called permanent gases varies appreciably, so that there is a slight difference in the position of the absolute zero according to the gas under experiment.

The assumed zero-point is the absolute zero of the scale, and is approximately $493.2^{\circ} - 32^{\circ} = 461.2^{\circ}\text{F}$. below the ordinary zero Fahrenheit.

This is shown graphically in Fig. 1.

The Specific Heat of a substance is its capacity for heat. As usually expressed, it is the quantity of heat, stated in thermal units, which must be transferred to or taken from a unit of weight of a given substance, in order to raise or lower its temperature one degree at a certain specific temperature.

Specific heats are not constant for solids and liquids. They become greater as the temperature increases; and, further, the greater the coefficient of expansion of a substance, the greater will be the increase in the specific heat.

The specific heats of the perfect gases remain constant, as far as temperature or density is concerned, so that equal increments

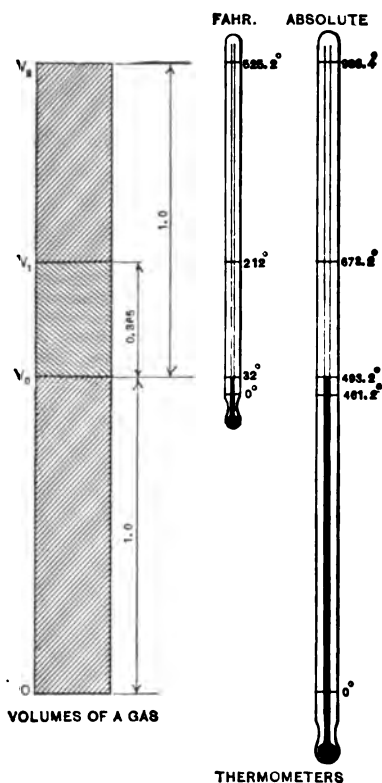


FIG. 1.—The Absolute Zero.

of temperature correspond to equal quantities of heat. Hence we may infer that at the absolute zero gaseous bodies become entirely destitute of the condition called "heat."

It was shown by LaPlace that there were two kinds of specific heats for gases, one corresponding to constant pressure and one to constant volume.

When a gas is heated at constant pressure, the heat taken in

is $K_p(\tau_2 - \tau_1)$ and the work done is $P(V_2 - V_1) = c(\tau_2 - \tau_1)$. The difference is the amount of increase of internal energy, or

$$(K_p - c)(\tau_2 - \tau_1).$$

When a gas is heated at constant volume, no external work is done. Therefore the specific heat at constant volume is always less than that at constant pressure. In this case the heat taken in is $K_v(\tau_2 - \tau_1)$, which represents the increase of internal energy.

Equating these values,

$$K_v(\tau_2 - \tau_1) = (K_p - c)(\tau_2 - \tau_1), \text{ or } K_v = K_p - c.$$

The ratio of $\frac{K_p}{K_v}$ is usually denoted by γ .

TABLE I
THE SPECIFIC HEAT OF A FEW SUBSTANCES

Substance.	Weight in Pounds of 1 Cu. Ft. under 1 Atmosphere		Specific Heat.	
	at 32° F.	at 60° F.	K_v . Constant Volume.	K_p . Constant Pressure.
Water at maximum density.....	62.425	62.367	1.000	
Sea-water.....	64.090	
Ice.....	57.500	0.504	
Anhydrous ammonia.....	0.0482	0.393	0.508
Air.....	0.0807	0.0764	0.1697	0.2375
Oxygen.....	0.0892	0.0844	0.1542	0.2175
Hydrogen.....	0.0056	0.0053	2.4177	3.4090
Nitrogen.....	0.0782	0.0740	0.1729	0.2438
Gaseous steam.....	0.0502	0.370	0.480
Carbon monoxide gas.....	0.0782	0.0740	0.1733	0.2426
Carbon dioxide gas.....	0.1227	0.1161	0.1692	0.2169

Note.—Authorities differ as to these figures.

From the definition, the specific heat of water at its maximum density is unity. For other densities within the commercial range the difference is so slight that for all practical calculations it may be taken as constant.

Latent Heat. When a body changes its state a certain amount of heat is either taken in or given out. This exchange of heat is necessary in order to effect such change of state, and under like circumstances is always the same in amount. Thus, suppose a given amount of water is at 60° F. with a normal barometric pressure. Heat the water and its temperature will rise until it has

become 212° F. At this point increase of temperature will cease, but the water will continue to absorb heat and commence to generate steam, which process will continue until all the water has been vaporized. During the change of state each pound of water will absorb 965.7 B. T. U. This disappearance of heat represents work done on the particles composing the water as they are moved farther away from each other against the molecular attraction.

Had the pressure been greater than one atmosphere, then the water would not have boiled until the temperature had been raised more than 212° F., and also less than 965.7 B. T. U. would have been required per pound to effect the change of state. The boiling-temperatures and the heat absorbed to change the state are constant for their corresponding pressures.

The converse of the above is also true.

The heat thus absorbed or given out is called the "Latent Heat," in order to distinguish it from the "Sensible Heat," which is the heat necessary to change a body's temperature.

Latent Heat may be defined as the quantity of heat which must be communicated to or taken from a body in a given state in order to convert it into another state without changing its temperature.

The most important cases are:

1. The latent heat of fusion, or the conversion of solids into liquids.

The following table gives the latent heat of fusion of a few substances, expressed in British thermal units per pound, under one atmosphere of pressure. These quantities change but little with variations of pressure.

TABLE II
LATENT HEAT OF FUSION OF A FEW SUBSTANCES

Ice.	144.0	Tin.	25.6
Cast iron.	233.0	Bismuth.	22.7
Zinc.	50.6	Lead.	9.6

2. The latent heat of evaporation, or the conversion of liquids into the gaseous state.

The following is a table of the latent heat of evaporation of a few substances when the pressure of the vapor is at one atmosphere, expressed in British thermal units. This latent heat decreases very perceptibly as the pressure increases.

TABLE III
LATENT HEAT OF EVAPORATION OF A FEW SUBSTANCES

Substance.	Boiling-point.	Latent Heat.
Water.....	212° F.	965.7
Anhydrous ammonia.....	-28.5° F.	572.7
Alcohol.....	172.2° F.	364.3
Ether.....	95° F.	162.8

3. The latent heat of expansion, or the heat which disappears in causing the volume of a body to increase under a constant pressure.

4. There are many chemical changes during which heat is generated or disappears.

For all work in connection with the design and management of steam-boilers the latent heat of evaporation is by far the most important.

As has already been stated, this latent heat varies in amount with the pressure under which the vaporization takes place, but is constant for the same pressure. The amount of latent heat required decreases as the pressure increases.

Furthermore, the same quantity of latent heat which was absorbed in the first place as latent heat of evaporation must be again given out by the body when it changes its state back from the vapor to the liquid; and this heat must be transferred to some other body and carried away, in order that this process of condensation may go on.

The following empirical formula represents with great accuracy the experiments of M. Regnault on the latent heat of evaporation of water. The latent heat of one pound of water in British thermal units is denoted by l , and any temperature Fahrenheit by T ; then

$$l = 1091.7 - 0.695\{T^\circ - 32^\circ\} - 0.000,000,103\{T^\circ - 39.1^\circ\}^2.$$

As this formula is rather complicated for easy use, it is sufficient for all practical work, when a steam-table is not at hand, to use the following form, thus:

$$l = 1092 - 0.7\{T^\circ - 32^\circ\} = 966 - 0.7\{T^\circ - 212^\circ\}.$$

Total Heat of Evaporation. The "total heat of evaporation" is a conventional phrase, used to denote the heat that is taken in by a substance when it is raised from some lower temperature and evaporated at a higher temperature. The heat necessary to raise the temperature from the lower temperature to that of evaporation is known as the "sensible heat."

The total heat of evaporation is then the sum of the sensible and latent heats.

M. Regnault found by experiment that the total heat increased for water at a uniform rate as the temperature of evaporation rises. He proposed the following empirical formula for calculating the total heat of evaporation for water, thus:

$$h = 1091.7 + 0.305\{T^{\circ} - 32^{\circ}\}.$$

In this expression h denotes the total heat of one pound of water, raised from the freezing-point to any temperature T° F.

As in most cases the lower fixed temperature is above that of melting ice, the total heat of evaporation will then be less than that given by the formula, by the amount of heat contained between 32° F. and such initial temperature.

Without causing an error of any practical moment, small fractions may be neglected, and also the specific heat of water may be taken as constant, at unity. Then, for all commercial purposes, whenever it be required to determine the total heat of evaporation from any temperature T_2° to another at T_1° , Regnault's formula may be simplified as follows:

$$h_{2,1} = 1092 + 0.3\{T_1^{\circ} - 32^{\circ}\} - \{T_2^{\circ} - 32^{\circ}\}.$$

Weight of Water. Rankine's empirical formula to calculate the weight of water at different temperatures, will often be found convenient when a water-table is not at hand.

The weight of one cubic foot of water at any temperature, T° , will be very nearly equivalent to the following expression:

$$\frac{2 \times 62.425}{\tau} \frac{500}{500 + \tau}$$

τ denoting absolute temperature corresponding to T° F.

Boiling. On the application of heat the temperature of water increases until it reaches the "boiling-point." The water will continue to absorb heat, although its temperature will remain constant. When sufficient latent heat has been absorbed in order to effect a change of state, the water will begin to boil. The temperature of the boiling-point increases with the pressure, but is always constant for its corresponding pressure.

Water does not boil from the surface, but bubbles of steam form throughout the mass and rise to the surface. This action is very violent in steam-boilers, and bubbles of steam rise so rapidly as often to carry considerable water in mechanical suspension into the steam. This action is called "priming," and it occurs most frequently in poorly designed boilers and in those that are forced beyond their capacity. It is also encouraged by use of dirty or greasy water.

The action of boiling is resisted when made to take place in glass vessels and in those formed of materials which attract water. In such vessels ebullition is not continuous. Ebullition, therefore, is not truly represented in small glass models, although many have been used by selling agents to illustrate some so-called faulty action in other makers' boilers.

Salt water or brine also resists ebullition, and the boiling-point for salt water is higher than that for fresh water under the same pressure.

The boiling-point for saturated brine under one atmosphere is 226° F. For each $\frac{1}{2}$ part by weight of salt which the water contains the boiling-point is raised about 1.2° F.

Average sea-water contains about $\frac{1}{2}$ of salt, but it varies somewhat in different parts of the globe. It is usual to speak of the quantity of salt contained as being in 32ds, although the quantity is often expressed as so many ounces to the gallon.

When sea-water is used in marine boilers the brine should not be allowed to get stronger than $\frac{2}{3}$ or $\frac{3}{4}$, and preference should be given to the lesser limit. Saturated brine contains about 30 per cent of salt, or nearly $\frac{1}{2}$.

The strength of the solution in the boiler should be tested at frequent intervals by blowing out a little water into a pail, and measuring its strength by a hydrometer or salinometer, the zero of which instrument is the floating mark in fresh water.

Salinometers may be bought, but in case of breakage can easily be replaced temporarily by using a vial weighted with shot so as to make it float in an upright position. Place the vial in pure water and scratch a mark for the zero-point. Then place in saturated brine and mark again. Divide the space between the marks into ten equal divisions, and each will represent approximately $\frac{1}{10}$ of salt.

Relative and Specific Volumes of Steam. The volume of any given portion of steam, compared to that of the water from which it was evaporated, is called the relative volume of steam for that corresponding pressure. The specific volume is the volume of steam generated from one pound of water.

Under one atmosphere or 14.7 pounds per square inch absolute, one cubic foot of water will occupy about 1642 cubic feet when converted into steam. Under ten atmospheres or 147 pounds, the steam would occupy nearly 189.7 cubic feet. From these two examples can be seen what an enormous expansive force there is in steam.

Steam-tables give the corresponding absolute pressures, temperatures, total heats of evaporation, weights and volumes.

Factor of Evaporation. The value of any fuel as a heat-generating agent is generally expressed in the "weight of water that it will evaporate per pound." But the temperature of the feed-water and the temperature or pressure at which evaporation takes place will greatly affect the quantity evaporated and therefore the *apparent value* of the fuel. In order to make all results comparable, it is customary to reduce the actual amount of water evaporated to that which would have been evaporated had the feed-water been supplied at a temperature of 212° and the evaporation taken place at 212°, that is, under one atmosphere of pressure.

This result is called "*the equivalent evaporation from and at 212°*," and the weight of water so found per pound of fuel is said to be "*the evaporative power of the fuel*."

To find this quantity, it is only necessary to determine the total heat of evaporation under the actual conditions by means of the formula, or from the steam-tables, and divide by 966, the latent heat of evaporation of water at 212°. The quotient will be a multiplier, by which the actual evaporation must be multiplied in order to get the equivalent quantity from and at 212°. This

multiplier is called the "*factor of evaporation.*" A convenient expression for determining this factor of evaporation is:

$$\text{Factor of Evaporation} = 1 + \frac{0.3\{T_1^\circ - 212^\circ\} + \{212^\circ - T_2^\circ\}}{966},$$

in which T_1° denotes the temperature of the steam and T_2° that of the feed-water.

TABLE IV
ABSOLUTE PRESSURES, BOILING-POINTS AND FACTORS OF EVAPORATION

Pressures, Absolute, per Square Inch.	Boiling-point.	Factors of Evaporation, Feed-water Temperatures.	
		50°.	104°.
14.7	212° F.	1.169	1.113
52.5	284° F.	1.190	1.136
90.0	320° F.	1.203	1.147
115.3	338° F.	1.208	1.152
146.0	356° F.	1.214	1.158
160.0	363.4° F.	1.216	1.160
200.0	381.7° F.	1.222	1.166
336.0	428° F.	1.235	1.179

Example. Assume the following data:

A boiler evaporates per hour, actual = 12,000 lbs. of water

Coal burnt per hour. = 1,400 lbs.

Actual water evaporated per pound of coal

$$\text{per hour} = \frac{12000}{1400} \dots\dots\dots = 8.57 \text{ lbs.}$$

Temperature of feed-water, T_2° = 104°

Temperature of steam at 120 lbs. gauge pressure, T_1° = 350°

Then

$$\text{Factor of evaporation} \dots\dots\dots = \frac{h_{2,1} = 1115.4}{966} = 1.154$$

and equivalent evaporation from and at 212°

$$\text{is } 8.57 \times 1.154 \dots\dots\dots = 9.89 \text{ lbs.}$$

CHAPTER II

COMBUSTION

General Conditions. Definition. Smoke. Coal-Gas. Marsh-Gas. Olefiant Gas. Air. Temperatures of Ignition. Laws of Avogadro. Requirements for Perfect Combustion. Products of Combustion. Composition of Gases from Combustion. Refuse. Loss of Unburned Coal in Ash-pit. Quantity of Air Required. Methods of Firing. Thickness of Fire. Heat of Combustion. Heating Power of a Fuel.

As the power developed by the steam-engine is derived from the form of energy called "Heat," and as this heat is obtained by the combustion of a fuel, it is essential that the principles involved and the natural laws relating thereto be clearly understood. Furthermore, since the engineer designs the engine to perform a certain amount of work at a high economy, there should be no deficiency of steam or want of heat, or no excess of steam or too great an expenditure of fuel. Unfortunately, many steam plants have given poor satisfaction, simply from want of care in the design of the furnace.

By combustion is meant "chemical union," and in general this union is productive of heat.

It is a union between a combustible or fuel and a supporter of combustion. This supporter of combustion, within the limits of this work, is the oxygen contained in the atmosphere.

The principal fuels are coal, wood, gas and oil.

The chief constituents of these fuels are carbon and hydrogen, but their characteristics and modes of entering into combustion are very different.

The carbon is reduced to carbon dioxide,* the hydrogen to water or steam, sulphur to sulphurous or sulphuric acid, and any other elements, commonly called impurities, to their respective oxides.

* Carbon dioxide is also known as carbon anhydride, and frequently, although erroneously, as carbonic acid.

A fresh charge of coal when thrown on a fire in an active state becomes a great absorbent of heat. This apparent loss of heat is utilized in volatilizing the bituminous portion, and is a very cooling process, due to the change of sensible into latent heat. While this generation of the gases is taking place the carbonaceous part remains black or at a low temperature, awaiting the proper time for it to burn.

If the bituminous portion be not utilized in the gaseous state for the production of heat, it becomes a total loss and were better absent, as in that case all the latent heat would have been available. It is due to this fact that the bituminous coals do not give such an intense heat as the anthracites.

The above reasoning will explain why firemen throw fresh coal into a furnace in order to temporarily cool it, as, for instance, when the engine suddenly stops, or for some other cause there is a lessened demand for steam.

In order to effect complete combustion, the particles composing the gaseous and carbonaceous portions of the fuel must be brought into contact with the oxygen of the air supplied. The great difficulty is the proper mixing of the gases. If all the carbon is burned to carbon dioxide, there must be an excess of air passing through the furnace. If all the carbon is not burned in the short time allowed with a powerful draft, due to a lack of mixture or to a deficiency of air, the carbon is wasted as carbon monoxide or half-burned carbon, or in vaporized carbon which is commonly called smoke.

If once smoke be produced, it will be a difficult matter to consume it. It is not so difficult to burn coal without producing smoke by a proper admixture of air, introduced in suitable proportions and in a manner to bring the particles of carbon in contact with the oxygen when at high temperature. This is the real result that should be accomplished. For ordinary practice it is then a misapplication of words to say "how smoke from coals may be burned." The more correct expression would be "burning coals without producing smoke."

No definite rule can be laid down for the admission of air so as to burn all kinds of coal without producing smoke, as each variety of coal has its peculiar qualities and as so much depends on the design of furnace, grate, nearness of heating surfaces and strength

of draft. What is desired is that the air shall be thoroughly mixed with the particles of fuel before the latter are too much cooled by contact with the boiler surfaces. For some bituminous coals, a supply of air admitted above the grate and also behind the bridge wall is often most desirable and necessary.

In the intense heat of a fiercely burning fire the bituminous coals are vaporized with such great rapidity, that it is practically impossible to burn all the gaseous portion before it flies to the chimney and passes beyond the reach of combustion. However, much may be accomplished by a regular firing of small quantities at a time in order to reduce the smoke nuisance. Some of the mechanical systems for firing have been very successful in this regard.

Of all the different kinds of furnaces designed for various purposes, the most persistent smoker is that of the steam-boiler. The reason is obvious, as there are no hot walls to radiate back the heat and thus aid combustion.

In some designs of boilers the furnace is enclosed in a fire-brick combustion-chamber, and the products are not admitted to the heating surfaces until after combustion has become more or less perfect. This arrangement has met with success in many instances, and could be carried much farther than it is.

The object of the boiler is to rob the fuel of its heat as quickly as possible; therefore every particle of gas and carbon that comes unburned into contact with the water surfaces is cooled below the temperature of perfect union, and must be drawn into the stack in its unburned condition, surplus of air or not, and must add to the volume of smoke.

Many smoke-consuming devices are advertised which claim a saving in fuel of from ten to twenty-five per cent. Authorities agree that the extreme loss due to smoke is less than five per cent; therefore if the advertised devices do save as much as they claim, they are misnamed. Instead of being "smoke-consumers," they should be called "heat-savers." *

When heat is applied to coal, the resulting combustion is effected as follows: *first*, the absorption of heat; *second*, the

* The engineer should be very careful not to place too much value on advertising matter, catalogue statements and the like, as they are apt to be misleading.

vaporization of the bituminous or hydrocarbon portion and its combustion; and *third*, the combustion of the solid or carbonaceous part. These actions are entirely separate and distinct, and must take place in the order as given. The hydrocarbon or bituminous portion consists of marsh-gas, olefiant gas, tar, pitch, naphtha, etc.

The flame is derived from the gaseous portion, and this explains why the soft or bituminous coals burn with more flame than the anthracites.

Coal-gas, taken by itself, is not inflammable, as a lighted taper placed in a jar of coal-gas will be extinguished. In order to consume it oxygen must be supplied, that is, the gas must be mixed with air. When this is done the gas will be consumed instantly, provided the proper temperature be present.

When a charge of fresh coal is thrown on a fire we cannot control the amount of gas that may be generated, but we can control the supply of air. Therefore it is essential, when soft coals are to be burned, that a certain amount of air be admitted in addition to the regular supply through the grate, during the periods of evolution of the gases. This can be accomplished by permitting air to enter above the grate, or directly into the combustion-chamber behind the bridge wall, or both. The quantity admitted should bear some suitable relation to the percentage of the hydrocarbons contained in the fuel. It is best in all cases to provide ample passages for the air, and then to admit the proper quantity as determined by trial and observation of the smoke produced.

In order to burn coal economically, it has been found necessary that an excess of air should be allowed to enter the furnace. If only the theoretical quantity be supplied, a large proportion of the carbon will either not be consumed or be only half burned to carbon monoxide (CO).

On the other hand, too great an excess, as well as a deficiency of air, is a detriment to the economical working of the furnace.

Much depends upon the design, especially with soft coals, for the requisite quantity may be supplied in a manner as not to be available; that is, the particles of oxygen may not come into contact with particles of carbon. In short, the air and particles of fuel may not mix, but rush to the chimney in "stream-lines."

Coal-gas is composed of hydrogen and carbon, and the principal unions are called:

Marsh Gas, or Carburetted Hydrogen, and
Olefiant Gas, or Bi-carburetted Hydrogen.

Marsh Gas consists of one atom of carbon and four of hydrogen, and the atomic weight is $12 + 4 = 16$. The chemical symbol is CH_4 .

Olefiant Gas consists of two atoms of carbon and four of hydrogen, and the atomic weight is $12 + 12 + 4 = 28$. The chemical symbol is C_2H_4 .

Atmospheric air is a mixture composed principally of oxygen and nitrogen. Neglecting moisture, impurities and decimals, the components are found mixed in the following average proportions:

Oxygen.	23	parts	by	weight
Nitrogen.	77	"	"	"

or

Oxygen.	21	"	"	volume
Nitrogen.	79	"	"	"

As the gases are driven off from the coal, due to the absorption of latent heat, they become mixed with the entering air. The result is that the hydrogen separates from the carbon and unites with the oxygen, forming water, or, more correctly speaking, vapor of water. The now free carbon also unites with oxygen in the formation of carbon dioxide (CO_2). Both of these combinations are productive of heat, thus making the process continuous.

After the hydrocarbon element has been separated in the form of gas, the coal remaining on the grate is composed chiefly of solid carbon. This is consumed by uniting with the oxygen in the air which passes up between the grate-bars. The union of the carbon with the oxygen may be in two proportions, forming bodies having very different characteristics.

If two atoms of oxygen unite with one atom of carbon, the result is carbon dioxide. But if one atom of oxygen only unites with one atom of carbon, the resulting formation is carbonic oxide or carbon monoxide. This monoxide may yet unite with another atom of oxygen, and when it does its combustion will be complete.

If, however, this carbon monoxide does not meet with the necessary oxygen while within the furnace, it will pass away only half burned.

The same result is attained in cases where a particle of free carbon meets with another of carbon dioxide, when two particles

of carbon monoxide will be formed. Should there still be lacking the necessary oxygen, then two particles of half-burned carbon will pass into the stack. This latter case is continually happening where the air has to pass upward through a thick mass of incandescent carbonaceous matter. The air entering through the hot grate-bars becomes heated, and its oxygen unites with the incandescent carbon, forming carbon dioxide, thereby producing heat which keeps the layer next to the grate in an incandescent state. This carbon dioxide, at a high temperature, then has to pass upward through the layer of solid carbonaceous matter above, and it takes up an additional portion of carbon, forming two particles of carbon monoxide ($\text{CO}_2 + \text{C} = 2\text{CO}$). See Fig. 2.

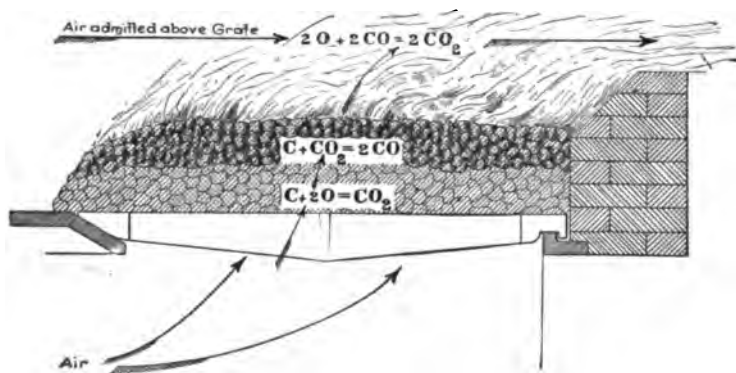


FIG. 2.—Combustion on Grate.

In this operation heat is absorbed, and there is also lost an extra portion of carbon, unless the monoxide meets with more oxygen to complete its combustion.

This illustrates why there always should be an excess of air passing through the furnace, and the possible advantage of having some air supplied above the grate or back of the bridge wall. Furthermore, the thickness of fire should be only sufficient to cover the grate properly and prevent too much air from passing. Thin fires have the disadvantage of burning through in spots, and are not liked by the firemen, who are thus compelled to maintain a close watch. Better results are obtained by using a thin fire, and supplying fresh charges at short, regular intervals, rather than by a complete spreading with heavier charges at longer intervals.

There is another peculiarity of this carbon monoxide, namely, that it will inflame at a lower temperature than the coal-gases (CH_4 and C_2H_2), due to its having already united with half its full capacity for oxygen. Consequently when the oxide has passed into the flues or has come into contact with the comparatively cool boiler surfaces, its temperature is often reduced below that required to burn the coal-gases, but is still hot enough to take up an extra portion of oxygen, which it frequently does on reaching the top of the chimney, where it becomes ignited on meeting the air. This explains the red flame so often seen at the top of chimneys, and necessarily the heat there generated is entirely lost for the purposes of the boiler.

The temperatures at which some of the physical and chemical changes take place when a fresh charge of coal is thrown on a fire are about as follows: *

(a) Previous to putting on a charge of coal the temperature of the bed of coals is from dull red heat (700°C. or 1292°F.) up to a bright white heat (1400°C. or 2552°F.) or even higher.

(b) The coal, when fired, is about 15°C. or 60°F. (temperature of the room). As soon as it reaches the fire-bed it begins to heat by conduction from the hot coals beneath. The hot gases, products of combustion of the coal beneath, also heat the new charge of coal.

(c) The heating of the coal causes the volatile matter to distil off. The amount distilled at any given temperature is unknown, but it is certain that traces of volatile combustible matters are given off as low as 110°C. (220°F.).

(d) At about 400°C. or 750°F. the coal reaches the temperature of ignition and burns to carbon dioxide.

(e) At about 600°C. or 1100°F. most of the gases given off by coal (hydrogen, marsh-gas and other volatile hydrocarbons) will ignite if oxygen be present.

(f) At 800°C. (1470°F.) the carbon dioxide, as soon as formed from the coal, will give up one atom of its oxygen to burn more coal, thus: $\text{CO}_2 + \text{C} = 2\text{CO}$. This carbonic oxide will burn back to carbon dioxide if mixed with oxygen at the necessary temperature, which is between 650° and 730°C. (1200° and 1350°F.).

(g) At about 1000°C. or 1832°F. the H_2O formed by the burning of the hydrogen in the volatile matter in the coal begins to dissociate.

* Steam Users' Association, Boston, Circular No. 9. R. S. Hale's Report on Efficiency of Combustion.

(h) At about 1000°C . or 1832°F . any carbon dioxide not previously burned to carbonic oxide begins to dissociate to carbonic oxide and oxygen.

(i) The various hydrocarbons which begin to be distilled at 110°C ., and possibly lower, undergo many changes, dissociations and breakings up at the various temperatures they pass through. So many of these are unknown that it is useless to state the few we do know.

Above 700°C . (1300°F .) both the hydrocarbons and the carbonic oxide will unite with oxygen if the latter be present and intimately mixed with them. If they do not burn, the tendency is always to break up into simpler and more volatile compounds as the temperature rises.

The above statements, however, give only the properties of the coal, and the chemical reactions it is capable of at the different temperatures it passes through. Its actual combustion depends on the supply of oxygen as well as on the condition of the coal at any given time. The oxygen is practically all supplied from the air, the amount of oxygen present in the coal being so small as to be of no present importance, even if it is not already in chemical combination with the carbon or hydrogen.

The temperatures at which some of the combinations mentioned take place were determined by Mallard and Le Chatelier, who published two articles in the *Annales des Mines*, Vol. IV, 1883, pp. 274, 379-559. In these experiments, mixtures of H and O, CO and O, and marsh-gas and O were placed in a closed chamber which was heated externally. They found that the hydrogen and oxygen ignited at 555°C ., the CO and O at 655°C ., and the marsh-gas at 650°C .

The results were found to be independent of the proportions of the gas in the mixture. It was also found that the presence of an inert gas such as N did not alter the results, with the exception that a large amount of CO_2 in the mixture of CO and O elevated the ignition temperature from 655°C . to 700°C .

With the H and CO mixed with O the combustion ensued immediately on exposure to the temperature of ignition, whereas with the CH_4 there was a lag in the ignition, time being required to ignite the gas after it was brought to the temperature of ignition. The above was taken from a brief account of the investigation given in the *Chemical Technology*, Vol. I, Groves and Thorp.

While discussing the subject of combustion it will be well to recall the laws of Avogadro (1811), which may be expressed thus: "The molecules of all gases, simple or compound, occupy equal

volumes; or equal volumes of all gases contain under similar conditions of temperature and pressure the same number of molecules." "The molecules of compound bodies in the gaseous state, with but few exceptions, occupy twice the volume of an atom of hydrogen." From this reasoning, the volume of CO is equal to twice the volume of CO_2 producing it. That is, when a particle of carbon burns into CO_2 and then meets another particle of carbon, the volume of the monoxide formed will be twice the volume of the original dioxide. This fact accounts for the loss in available heat.*

Furthermore, the production of carbon monoxide will require the same volume as if the carbon were burned to the dioxide, and while equally filling the flues and choking the draft, will only generate about one-third the heat.

The requirements for perfect combustion are a surplus of air, a thorough mixture of the fuel-particles with the oxygen in the air, and a high temperature. A furnace that fails to offer any or all of these conditions will not support perfect combustion.

The products of combustion are, therefore, carbon dioxide, carbon monoxide, vapor of water, the oxides of impurities in the fuel, oxygen, nitrogen and ash.

The composition of the gases from combustion may be found in almost any ratio. The following volumetric analyses will afford some idea of the ratio found. The last two are given on the authority of George H. Barrus, the last one being the products from Pocahontas (semi-bituminous) coal:

	Poor.		Average.	Excellent.
Carbon dioxide (CO_2).....	8.0%	9.0%	12.0%	15.1%
Oxygen (O)... ..	4.4	11.5	7.5	4.0
Carbon monoxide (CO)... ..	7.6	Trace	0.1	0.7
Nitrogen, vapor of water, etc., by difference.....	80.0	79.5	80.4	80.2
	100.0	100.0	100.0	100.0

These gas analyses can be made by the Orsat or some similar apparatus, by tapping the flue and extracting a measured volume

* See Rankine, Steam-engines, p. 270.

by means of a pressure-bottle, such as is used in a chemical laboratory, and a graduated burette. The sample is then forced in succession through three pipettes containing caustic potash, pyrogalllic acid and caustic potash, and cuprous chloride in hydrochloric acid, which will absorb respectively the carbon dioxide, the oxygen and the carbon monoxide. The loss of volume at each operation is measured in the burette.

From a gas analysis, the air-supply to the furnace can be closely calculated, as will be shown later.

The Refuse from a fuel is that portion which falls into the ash-pit and that carried off by the draft, consisting of ashes, unburnt or partially burnt fuel and cinders.

The following is from a report of R. S. Hale, Steam Users' Association, Boston, Circular No. 9:

"The amount of loss by unburned coal in the ash-pit depends on so many factors that it is impracticable to express it by any formula. A statement of the factors and a collection of examples must, therefore, suffice.

"(a) The loss by unburned coal in the ash-pit depends on the width of the opening in the grate-bars, and increases as the width increases.

"(b) The loss depends on the size of the coal, and increases as the size of the coal decreases.

"(c) The loss is probably greater for a non-caking than for a caking coal.

"(d) The loss probably increases as the amount of earthy matter in the coal increases, but not at the same ratio.

"(e)* The loss is less with a fan-blast than with a steam-blast.

"(f)* The loss is greater the more the fire is disturbed. This is especially noticeable in automatic stokers with moving grate-bars.

"When determining the amount of carbon or combustible in the refuse, taking the difference between the amount of refuse shown by the boiler test and the amount of earthy matter shown by analysis of a sample of the coal is not sufficient, for two reasons: *first*, the sampling of the coal may easily be in error; and *second*, a considerable amount of earthy matter is at times carried into the flues and even up the chimney."

* Report of Coal Waste Commission, Pa., 1893, p. 31.

TABLE V

LOSS BY UNBURNED COAL IN ASH-PIT

Remarks—Authority.	Per Cent Refuse.	Per Cent Combustible in Refuse.	Per Cent in Total Coal.
E. B. Coxe (Trans. N. E. Cotton Mfg. Assn., 1895), using his travelling grate, on small-sized anthracite coal.	10.05 23.70	18.68 11.92	2.2 2.7
W. H. Bryan (Trans. A. S. M. E., Vol. XVI, p. 773), using soft coal.	13.35 14.31	31.0 25.0	4.3 3.6
Pennsylvania coal, bars 1½ in. wide, 1 in. apart.	16.10	25.0	4.0
Other tests.	10.30	37.2	3.8
“ “	9.20	31.3	2.9
“ “	18.50	29.3	5.4
“ “ with a mechanical stoker.	13.61	67.8	9.2
“ “ “ “ “ “	18.70	67.2	12.6
Arkansas State Geological Survey Report, 1888, Vol. III, p. 73. Pittsburg coal.	8.10	26.0	2.1
Ditto. Arkansas coal.	10.30	30.0	3.1
“ “ “ “ “ “	40.00	83.0	33.2
“ “ “ “ “ “	14.00	51.4	7.2
Dampfessel Revision Verein Berlin Geschäfts Bericht, 1895, p. 79. Coal-dust.	4.8	50.0	2.4

The quantity of air required for the complete combustion of the principal elements of a fuel may be determined as follows:

CARBURETTED HYDROGEN—MARSH-GAS— CH_4 .

Before Combustion.			After Combustion.	
Weight.	Atoms.	Weight.	Weight.	
CH_4 16	1 Carbon	12		44 Carbonic Acid
	1 Hydrogen	1		
	1 Hydrogen	1		
	1 Hydrogen	1		
	1 Hydrogen	1		
Air 278+	1 Oxygen	16		18 Water
	1 Oxygen	16		
	1 Oxygen	16		
	1 Oxygen	16		
	1 Oxygen	16		
	15.3 Nitrogen	214+		214+ Free Nitrogen
294+		294+	214+	
			294+	

From this diagram it will be noted that for every 16 parts by weight of marsh-gas 278+ parts of air are required for complete combustion; or for every one pound of marsh-gas 17.4 pounds of air are required.

BI-CARBURETTED HYDROGEN—OLEFIANT GAS— C_2H_4 .

Before Combustion.			After Combustion.	
Weight.	Atoms.	Weight.	Weight.	
C_2H_4 28	1 Carbon	12		
	1 Carbon	12		
	1 Hydrogen	1		
	1 Hydrogen	1		
	1 Hydrogen	1		
Air 417+	1 Hydrogen	1		
	1 Oxygen	16		
	1 Oxygen	16		
	1 Oxygen	16		
	1 Oxygen	16		
	1 Oxygen	16		
	1 Oxygen	16		
	23 Nitrogen	321+		321+ Free Nitrogen
445+		445+		445+
			44	Carbonic Acid
			44	Carbonic Acid
			18	Water
			18	Water

From this diagram it will be noted that for every 28 parts by weight of olefiant gas, 417+ parts of air are required for complete combustion; or for one pound of olefiant gas 14.9 pounds of air are required.

Air Required to Burn One Pound of Carbon. If the combustion be perfect, the air required to burn one pound of carbon will have to be sufficient to change the carbon into carbon dioxide. The composition of the dioxide is, by weight, 12 parts carbon and 32 parts oxygen, or one part carbon and 2.67 parts oxygen.

Also there are, by weight, 0.23 parts of oxygen in one part of air. Therefore one pound of carbon will require

$$0.23 : 1 :: 2.67 : x = \frac{2.67}{0.23} = 11.61 \text{ lbs. of air.}$$

If the carbon be imperfectly burned, it will be changed to the monoxide; and the composition of the oxide is, by weight, 12 parts carbon and 16 parts oxygen, or one part carbon and 1.33 parts oxygen.

Therefore, for imperfect combustion, one pound of carbon will require

$$0.23 : 1 :: 1.33 : x = \frac{1.33}{0.23} = 5.78 \text{ lbs. of air.}$$

Air Required to Burn One Pound of Hydrogen. When hydrogen is burned it forms a union with oxygen, the product of which is water. The relative weights of the combining volumes are in

the ratio of two parts hydrogen to sixteen parts oxygen, or one part hydrogen to eight parts oxygen, making nine parts water.

Therefore one pound of hydrogen will require

$$0.23 : 1 :: 8 : x = \frac{8}{0.23} = 34.78 \text{ lbs. of air.}$$

The Volume of Air Required for Combustion. Since it requires 11.61 pounds of air to burn one pound of carbon, and since the volume of one pound of dry air at 62° F., with a barometric pressure of 29.92 inches of mercury, is 13.141 cubic feet, the volume of air required for combustion at stated temperature and pressure is

$$11.61 \times 13.141 = 152.56 \text{ cubic feet per pound.}$$

By a similar course of reasoning, the volume required for the combustion of hydrogen is

$$34.78 \times 13.141 = 457.04 \text{ cubic feet per pound.}$$

The following formula of Dulong is convenient for determining the theoretical quantity of air that is required for the combustion of any fuel whose composition is known.

Let C, H and O denote respectively the weight of carbon, hydrogen and oxygen in the fuel; and W and V the weight and volume of air required. Other ingredients may be neglected, as they have but a slight effect on the result. Then

$$W = 11.61C + 34.78\left(H - \frac{O}{8}\right), \text{ or}$$

$$W = 12C + 35\left(H - \frac{O}{8}\right), \text{ nearly; and}$$

$$V = 152.56C + 457.04\left(H - \frac{O}{8}\right), \text{ or}$$

$$V = 153C + 457\left(H - \frac{O}{8}\right), \text{ nearly.}$$

The value of W per pound is about 12 for anthracite and good bituminous coals, 6 for wood, and 11 for charcoal.

It is found impossible in practice to obtain complete combustion unless the air supplied to the furnace be in excess of that

theoretically required. Experience dictates that for ordinary natural draft nearly twice the theoretical quantity of air should be admitted, or about 24 pounds per pound of coal. With mechanical drafts and with natural drafts when the mixing effects are strong and positive, the excess of air may be considerably reduced.

The volume of air-supply per pound of coal, in ordinary factory practice, with natural draft is about 300 cubic feet; and may be as low as 200 cubic feet when the mixing effect is strong.

The actual volume may be estimated by using an anemometer, or may be closely calculated from a gas analysis. This calculation is best illustrated by an example.

Take the gas analysis, marked average, in a previous paragraph, and consider the percentages of volume as cubic feet in one hundred of gas. The weights can be determined from the densities given in Table I.

	Vols.	Density.	Weights.
For CO ₂	12.0	$\times 0.1227$	$= 1.47240$
“ O.....	7.5	$\times 0.0892$	$= 0.66900$
“ CO.....	0.1	$\times 0.0782$	$= 0.00782$

These weights can be subdivided into those of their constituents; thus the CO₂ contains by weight $\frac{8}{11}$ of carbon and $\frac{8}{11}$ of oxygen, and the CO, $\frac{3}{4}$ of carbon and $\frac{1}{4}$ of oxygen.

$\frac{8}{11} \times 1.47240 = 1.07084$	$\frac{8}{11} \times 1.47240 = 0.40156$
$\frac{3}{4} \times 0.00782 = 0.00446$	$\frac{3}{4} \times 0.00782 = 0.00335$
0.66900	
Pounds of oxygen, . . . 1.74430	Pounds of carbon, . . . 0.40491

Therefore the oxygen per pound of carbon is $\frac{1.74430}{0.40491} = 4.30$ lbs.
Again, since air contains 0.23 parts of oxygen, the air per pound of carbon is $\frac{4.30}{0.23} = 18.7$ lbs.

As above reasoning assumes that all the surplus air is charged to the carbon, the final result will have to be increased by the theoretical amount necessary to burn the hydrogen.

Assume that the analysis of the coal was: carbon 87%, hydro-

gen 2%, oxygen 3% and ash 8%. Then the air in pounds supplied will be:

$$\text{For carbon. } 0.87 \times 18.7 = 16.27$$

$$\text{" hydrogen. } 35 \left(0.02 - \frac{0.03}{8} \right) = 0.57$$

$$\text{Air per pound of coal. } = 16.84 \text{ lbs.}$$

The theoretical amount of air for combustion would have been $W = 12 \times 0.87 + 35 \left(0.02 - \frac{0.03}{8} \right) = 11.01$ lbs., and therefore the surplus was 5.83 lbs., or about 53 per cent.

The Methods for Charging coal are known as the alternate, spreading and coking firings, in accordance with the way in which the fuel is spread upon the grate.

(a) The Alternate Method consists of charging the fresh coal on one side of the fire at a time, so that the gases evolved can be burned by the excess of air passing through the other side, which is at a bright heat. With boilers having two or more furnaces and a common combustion-chamber this is practically accomplished by firing only one furnace at a time. This method operates most effectively when the flow of the gases is such as to produce a thorough mixture.

(b) The Spreading Method consists of charging a thin layer of coal over the whole grate at each firing. When the gases have to rise vertically this method is generally considered better than the alternate method. It may be modified by sprinkling the charge in patches instead of covering the whole surface. This latter method is better than a complete spreading, and with the ordinary firemen will give more economical results.

(c) The Coking Method consists of charging the fresh coal on the dead plate at the front of the fire, and pushing back the coked fuel to make room for the new charge. This method is only advantageous when the gases evolved pass over the bright part of the fire. It is of little or no use when the gases have to rise vertically. It is also a difficult method and requires a fairly good fireman to make it effective.

Much depends on the fuel and the furnace design, but in general the spreading method is the best and then the alternate. In

any case the charges should be in small quantities at frequent intervals.

The Thickness of Fire varies from about three inches to about sixteen inches. Fine sizes of coal must be used in thin fires, as they pack so close as greatly to restrict the draft. A thick fire requires more air admitted above the grate to consume the carbon monoxide than does a thin fire. It is best to use as thin a fire as the coal will permit. Thick fires being more easy to handle are preferred by firemen. Thin fires require closer attention to prevent holes being burned in spots, and less readily respond to sudden calls for steam.

A thin fire has the simplicity of letting all the air required pass through the grate, which is thus warmed and mixed to best advantage. When the gases rise vertically it is very difficult, unless complicated methods be adopted, properly to admit air above the grate and accomplish a complete mixture.

A Jet of Steam admitted above or below the fire has no corresponding advantage unless it be strong enough to produce an artificial draft. A small jet is an uneconomical method in the use of steam. The steam may produce water-gas, but no additional heat is produced thereby. It may prevent clinkering with some of the cheap fuels, and may reduce the smoke by a process of collection of the particles, but no economy is effected over the cost of the steam used. For similar reasons water is sometimes put in the ash-pit or the fuel purposely wet. Both these latter methods are sources of direct loss.

The conclusions drawn by R. S. Hale * are: That ordinary firing is apt to give 10 to 20 per cent worse results than the best skilled firing, the low results being caused by using too much air and by getting poor combustion.

That it is easier for firemen to get better results in some boiler-furnaces than others, but that this difference becomes large only with poor soft coal.

That many but not all of the patent devices (down-draft grates, stokers, etc.) in common use will with moderately skilled firemen give better results than those obtained by ordinary firemen in ordinary furnaces.

* Steam Users' Circular No. 9.

That it is probable, but not proved, that ordinary firemen can get better results from these devices than can ordinary firemen on ordinary grates.

Heat of Combustion. The heat produced by the combustion of one pound of various substances is given in the following table in British heat-units:

TABLE VI
TOTAL HEATS OF COMBUSTION

Hydrogen gas.....	62,032
Carbon to carbon dioxide.....	14,500
Carbon to carbon monoxide.....	4,400
Carbon monoxide to carbon dioxide.....	4,330
Olefiant gas.....	21,344
Liquid hydrocarbons vary in proportion to weight from	19,000
	to 22,600
Charcoal, wood.....	13,500
" peat.....	11,600
Wood, dry.....	average 7,800
" 20% moisture.....	6,500
Peat, dry.....	average 9,950
" 25% moisture.....	7,000
Coal, anthracite, best qualities.....	about 15,000
" " ordinary.....	" 13,000
" bituminous, dry.....	" 14,000
" cannel.....	" 15,000
" ordinary poor grades.....	" 10,000

These figures are slightly altered by different authors. The above list may fairly be taken as an average.

The heating-power of any fuel, that is, its total heat of combustion, is determined by calculation or by actual measurement in a calorimeter. This quantity is the sum of the amounts of heat generated by the combustion of the unoxidized carbon and hydrogen contained in the fuel, less the heat required in the evaporation and volatilization of those constituents which become gaseous at the temperatures resulting from the combustion of the first-named constituents.

The heating-power may be expressed for nearly all practical purposes with sufficient accuracy by the formula of MM. Favre and Silbermann, as follows:

$$\text{Total heat of combustion in B. T. U.} = 14,500C + 62,032\left(H - \frac{O}{8}\right),$$

in which C, H and O represent the proportions by weight of carbon, hydrogen and oxygen contained in the fuel. One-eighth of

the weight of oxygen is subtracted from the hydrogen, because oxygen and hydrogen unite in that proportion to form water, and when present in that proportion are useless for the production of heat.

Later experiments show that a closer agreement to calorimetric results is obtained by modifying the coefficients, thus:

$$\text{Total heat} = 14,600C + 62,000\left(H - \frac{O}{8}\right) + 4000S,$$

in which C, H, O and S represent the proportions of carbon, hydrogen, oxygen and sulphur.

An approximate formula, which will be found convenient for agents purchasing coals, because the percentages of ash and moisture are easily obtainable, is in use in the following form:

$$\begin{aligned} \text{Total heat of combustion} = h = \\ 154.8\{100 - (\text{per cent of ash} + \text{per cent of moisture})\}. \end{aligned}$$

CHAPTER III

FUELS

Coal. Classification. Anthracite. Semi-anthracite. Semi-bituminous. Bituminous. Dry Bituminous. Bituminous Caking. Long-flaming Bituminous. Lignite. Size of Coal. Culm. Weight of Coal. Peat or Turf. Wood. Coke and Charcoal. Miscellaneous Fuels. Sawdust. Straw. Bagasse. Protection from Weather. Chemical Composition of Coals. Liquid Fuels. Gaseous Fuels.

EVERY form of fuel is especially suitable for particular purposes and conditions, whether it be in a solid, a liquid or a gaseous state.

In making his selection, the engineer must choose the one best adapted to the work in hand, taking into consideration all the circumstances that may affect its use. The selection is often dependent on the ease or difficulty with which it can be procured, as well as its cost. The cost frequently prevents the best fuel from being used; for, although less may be required, still the price may be so high as to render a larger quantity of some cheaper grade more economical.

Having made a selection of kind and quality, the engineer then designs the boilers and furnaces to suit.

Coal. All coals are of vegetable origin, being the long-decayed product of ancient forests. Although coal has undergone a complete change from its original state, its chemical composition often is little altered. However, coal is sometimes found so mixed with earthy matters that its value as a fuel is entirely lost.

When burned, the organic matter is resolved into its various component parts, consisting of carbon, hydrogen and oxygen, combined in formation of various substances, as carbon, tar, ammonia, benzole, naphtha, paraffine, the coal-gases and coke; while the inorganic matter remains as ash, consisting chiefly of the silicates.

It is difficult, if not impossible, to distinguish the coals by name and to classify all varieties under proper headings or subdivisions, since they are found in all forms intermediate between that of recent vegetable growth to that of the perfectly mineralized state.

The classification of M. L. Gruner is as follows: *

Name of Fuel.	Ratio $\frac{O}{H}$.	Proportion of Coke or Charcoal Yielded by the Dry Pure Fuel.
Anthracite coals.	1 to 0.75	.90 to .92
Bituminous coals.	4 to 1	.50 to .90
Lignite or brown coals.	5	.40 to .50
Peat and fossil fuel.	6 to 5	.35 to .40
Wood (cellulose and encasing matter).	7	.30 to .35
Pure cellulose.	8	.28 to .30

A general classification as proposed by William Kent, based on a method by Prof. Persifer Frazer, is as follows:

The "fuel ratio" or "carbon ratio" is the ratio the fixed carbon bears to the volatile hydrocarbons. This arrangement considers only the fuel constituents and disregards the accidental impurities, such as sulphur, earthy matter and moisture.

Kind of Fuel.	Fixed Carbon, Per Cent.	Volatile Hydrocarbons, Per Cent.	Fuel Ratio $\frac{C}{V.H.C.}$
1. Hard dry anthracite. ...	100 to 92.31	0 to 7.69	100 to 12
2. Semi-anthracite.	92.31 to 87.50	7.69 to 12.50	12 to 7
3. Semi-bituminous.	87.50 to 75.00	12.50 to 25.00	7 to 3
4. Bituminous.	75.00 to 0	25.00 to 100	3 to 0

Rankine classifies the coals thus: Anthracite, Semi-bituminous, Bituminous, Long Flaming or Cannel, and Lignite or Brown Coal.

Anthracite. This is coal in its most perfect form, and is found in the oldest carboniferous strata. Its qualities are hardness and compactness. It is intermediate in color between jet-black and plumbago. It is amorphous and vitreous, and has a specific gravity of from 1.4 to 1.6. While having a high calorific value, it is difficult to "fire" and, when fired, to keep lighted. Many sam-

* Engineering and Mining Journal, 25 July, 1874.

ples split into small pieces when heated, which cause a considerable loss by their falling into the ash-pit through the grate-bars before being burned. Being free from the hydrocarbons, it burns with little flame and produces but a small amount of smoke.

The percentage of refuse varies considerably, and the loss due to "splitting" often increases the actual amount to nearly double. According to size as well as quality, the refuse varies from five to over sixteen per cent, the coarser sizes giving the least amount.

Semi-anthracite. This is a coal situated between the pure anthracite and the semi-bituminous. It is less amorphous and more lamellar than anthracite, is less hard and burns more freely. It can generally be distinguished by its tendency to soil the hands, while pure anthracite will not.

Semi-bituminous. This is the next grade toward bituminous coal. It burns still freer, contains more volatile hydrocarbon, and is a valuable steaming coal.

Bituminous. This grade is very extensive, and contains some very valuable varieties. All the bituminous coals need firing with care to prevent smoke and clinkers. Some of the grades have a very high calorific value and are much used for steam purposes.

The class is usually divided into three grades:

(a) *Dry Bituminous.* This coal has a specific gravity between 1.25 and 1.40; a color nearly black, with a resinous lustre. It burns freely and kindles with much less difficulty than the anthracites. It is hard, but weak and splintery. It gives a moderate amount of flame and but little smoke.

(b) *Bituminous Caking.* This coal has a specific gravity of about 1.25 and contains less carbon and more of the hydrocarbons than the former class. Its color is less black and more resinous, and there is less tendency to splinter. As the hydrocarbons are driven off, this coal breaks into smaller pieces which become pasty and finally unite into large solid masses. Unless frequently broken up these masses check the draft. The flame is of a yellowish color. It is a valuable coal for the manufacture of gas and for burning in open grates.

(c) *Long-flaming Bituminous.* This coal is similar in many respects to the latter class, but contains less carbon and more hydrogen. It is free-burning with a long yellowish flame, and has a strong tendency to cake or form clinkers.

Lignite. This variety is sometimes called "brown coal" on account of its color. It is really coal from the more recent geological formations, and is therefore less perfect. Its specific gravity varies from 1.10 to 1.25, the heavier samples containing the greatest percentage of earthy matters. It kindles with ease and burns freely, and is therefore consumed rapidly. Its structure is woody; it is lustreless, contains a large amount of water, and even when dried will again readily absorb large quantities.

It forms a poor fuel when judged by its evaporating power, but is largely used in certain localities owing to its cheap cost.

Size of Coal. The trade distinguishes the sizes by certain names, which refer to the dimensions of the lumps or pieces and not to the grade.

As the bituminous coals are not sold according to size they are known only as "run of mine" or "screened." The anthracites and semi-anthracites are, however, sold under trade names, which vary somewhat as regards the dimensions in different localities. The "mesh" of the screens over or through which the coal passes while being separated into sizes will not be found to differ materially from the accompanying list. For sizes above "broken coal" bars are generally employed instead of screens. Each coal company does not always sell all the listed sizes, but more often confines itself to some special ones. The greatest demand, as measured by tonnage, is for broken, pea and buckwheat.

LIST OF SIZES OF ANTHRACITE COAL

Trade Name.	Size of Screen.	
	Over.	Through.
Run of mine.	All sizes mixed	
Lump.	5 inches	
Furnace lump.	Selected for blast-furnace	
Steamboat lump.	4½ inches	7 inches
Broken or grate.	2½ "	4½ "
Egg.	2½ "	2½ "
Large stove or stove No. 1.	1½ "	2½ "
Small stove, stove No. 2, or range.	1½ "	1½ "
Chestnut.	1½ "	1½ "
Pea or nut.	1½ "	1½ "
Buckwheat, or buckwheat No. 1.	1½ "	1½ "
Rice, or buckwheat No. 2.	1½ "	1½ "
Barley, or birdseye.	1½ "	1½ "
Culm.	Dust	

Culm. This is the name given to the refuse dust at the coal-mines. It is sometimes called "slack" or "breeze." It can be bought at the mines at very low rates, as it is difficult to transport, being subject to heavy loss due to its fineness. Its use is, therefore, local. Efforts have been made to compress this dust into briquettes, but so far its adoption has been but limited. Possibly a successful method may yet be invented.

On account of its fineness, culm cannot be burned on the ordinary grate. There are various methods of burning it, all being based on the principle of blowing the dust into the furnace with the requisite quantity of air. It then burns much like gas. Sometimes a small grate is used on which there is the usual fire, for the purpose of igniting the dust-blast in case the flame be extinguished. For best results the culm should be first pulverized to a fine powder before being blown into the furnace.

TABLE VII
WEIGHT PER CUBIC FOOT OF VARIOUS COALS *

	Weight per Cubic Foot, Pounds.	Cubic Feet per Ton, 2000 Pounds.
Lehigh lump.	55.26	36.19
" cupola.	55.52	36.02
" broken.	56.85	35.18
" egg.	57.74	34.63
" stove.	58.15	34.39
" nut.	58.26	34.32
" pea.	53.18	37.60
" buckwheat.	54.04	37.01
" dust.	57.25	34.98
Free-burning egg.	56.07	35.67
" stove.	56.33	35.50
" nut.	56.88	35.16
Pittsburg.	46.48	43.03
Illinois.	47.22	42.35
Cornellsville coke.	26.30	76.04
Hocking.	49.30	40.56
Indiana block.	43.85	45.61
Erie.	48.07	41.61
Ohio canal.	49.18	40.66

When Buying Coals it is well to remember the following suggestions:

(a) The heating power per pound of combustible of the com-

* Extract from bulletin of Anthracite Coal Operators' Association for November, 1897.

bustible portion is about constant; and more attention should be given to the percentage of earthy matter contained than to the calorific power per pound of coal.

(b) The percentage of earthy matter appears to increase by about $1\frac{1}{2}$ per cent for each size of coal, as it becomes smaller, but the price often diminishes in a greater ratio.

(c) The amount of refuse is always much in excess of the earthy matter as reported by analysis.

(d) With anthracites the best qualities are indicated by the sharpest angles and the brightest appearance. If the coal is dull and shows seams and cracks, it will split into small fragments in the heat of the furnace and will not prove economical.

(e) Bituminous coals should be avoided which show fractures with whitish films or rusty stains as being indications of the presence of sulphur and pyrites.

Peat or Turf. This fuel is obtained from bogs and similar places, and consists of the woody roots of plants mixed with earthy matters. It contains large percentages of moisture, and even when dried will remain so only under great care. In some localities it is pressed into blocks or briquettes of convenient size. Its commercial use is very limited for steam-raising purposes. When air-dried it contains about 25 to 30 per cent of moisture, ash from 3 to 12 per cent, and has a specific gravity of 0.4 to 0.5 in the ordinary state.

Wood. This fuel is largely used in certain districts. When freshly cut it contains about 40 per cent of moisture, depending on the kind. After being air-dried for 8 or 10 months it will still contain at least 20 per cent. When dried it contains on the average 50 per cent of carbon and 50 per cent of oxygen, hydrogen, etc. The specific gravity varies from about 0.3 to 1.2, and the amount of ash from 0.5 to 6 per cent. The lighter varieties burn the most readily, while the denser kinds give the most heat and burn longest.

The heating power of one cord of hard wood is about equal to one ton of average anthracite; while one cord of soft wood is a little less than half a ton. The heating value of the different species is about the same when compared by weight, that is, one pound of hard wood is about equal to one pound of soft wood. The heating power of dry wood used as a fuel is usually assumed as

equal to 0.4 that of average soft coal of same weight; that is, 2½ pounds of wood are equivalent to one pound of coal.

Coke and Charcoal. These fuels are made by evaporating the volatile constituents from coal and wood respectively. They both give a very hot fire, but on account of expense are not used commercially for steam-making.

Miscellaneous Fuels. Sawdust, Straw, Bagasse. Sawdust is a favorite fuel in sawmills and in their vicinity, as when allowed to collect it becomes a source of danger from fire. It absorbs moisture very quickly, more so than the wood from which it was produced, on account of its increased surface. It has the same heating power as the original wood. It requires a large supply of air to properly consume it, and therefore the furnace and combustion-chamber should be given liberal proportions.

Straw as a fuel is only used when it becomes the cheapest method to get rid of it. It has a heating power varying from about 5000 to 6000 heat-units per pound. Its combustion is not unlike that of wood shavings.

Bagasse or megass is refuse sugar-cane and is used as a fuel on the sugar-plantations. Owing to the woody fibre, the sugar and other combustibles contained, it gives off a great amount of heat. As single crushing of the cane extracts about 66 per cent of the sugar juice, and double crushing about 72 per cent, the green bagasse consists approximately of:

	Single-crushed.	Double-crushed.
Woody fibre	37%	45%
Sugar	10	9
Water	53	46

When consumed at a high temperature, the oxygen contained is nearly sufficient to satisfy the carbon and hydrogen, so that little surplus air is required. Under favorable conditions about 1.1 pounds of bagasse are equivalent to 1 pound of Welsh coal.

The furnace* is constructed of brick, independent of the boilers, and the crushed cane is continuously fed by a belt conveyor into a hopper, often arranged so as automatically to control the amount. A form of bagasse furnace is shown in Fig. 3 and arranged to heat two sets of boilers. Another form of furnace is shown in Fig. 4.

* See "Megass Furnaces," Proceedings, Inst. C. E., Vol. 167.

STEAM-BOILERS

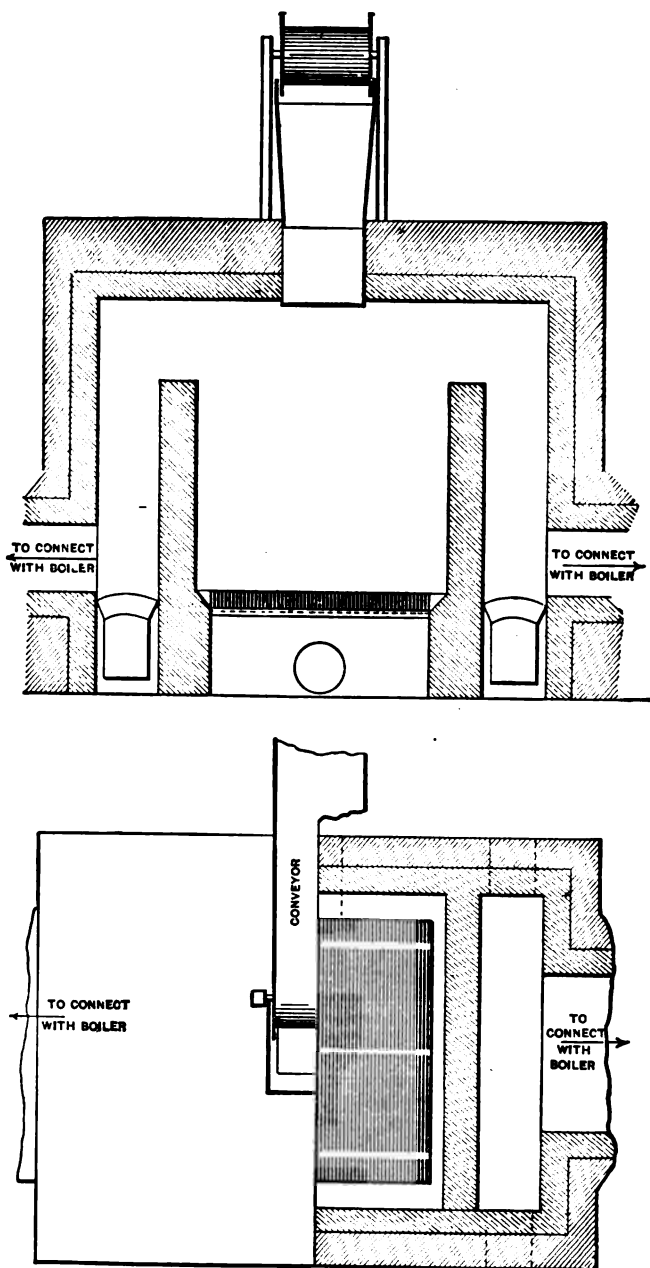
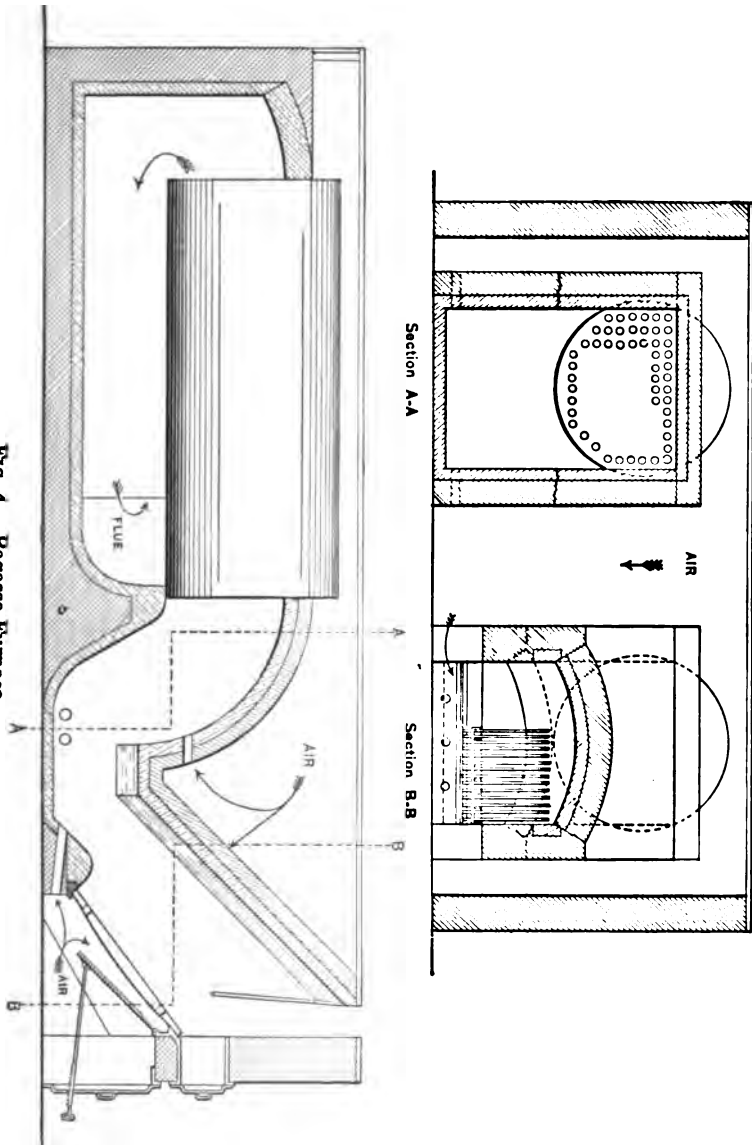


FIG. 3.—Bagasse Furnace—Stillman Type.



Protection from Weather.—All the solid fuels should be properly housed from the weather. When exposed they absorb moisture in greater or less amounts, the evaporation of which causes a loss. All the coals, but especially the bituminous grades, undergo a waste when exposed, due to a slow absorption of oxygen from the atmosphere, which reduces their heating power. The saving by proper housing may be more than offset by the interest on the cost of a building.

Coals which contain the compounds of sulphur and iron are liable, especially when wet, to a rapid oxidation, the generation of considerable heat and finally spontaneous combustion. Such coals should be stored in dry places, well ventilated. This applies especially to the coal-bunkers on ships.

Owing to the large space required, it often becomes impossible to house wood when stacked for fuel purposes. The wood is usually cut into "cord lengths," that is, four feet long, and should be neatly and evenly piled so as to expose the sides of the pile to the sun as well as to have its direction at right angles to the prevailing winds, in order that the air may pass through the pile and assist in drying. All the pieces should be so laid as to shed rain, and all outside pieces or slabs be placed on the top in the nature of a roofing. The fire-room should be designed sufficiently large, so as to accommodate two stacks of wood, one for immediate use and the other for drying while awaiting its turn.

Chemical Composition of Coals. Coals differ widely as judged by their chemical analysis. It is not within the scope of this work to treat this subject fully, but reference should be made to works on the subject. Kent's "Handbook for Mechanical Engineers" gives a number of analyses of many different kinds of coal.

The principal difficulty in analyzing coal is to obtain an average sample. For details of an approved method, refer to the Code of Steam-boiler Trials as adopted by the American Society of Mechanical Engineers.

The heating value may be calculated by the Dulong formula, which with revised constants is:

$$\text{Total heat of combustion} = h = 14,600C + 62,000\left(H - \frac{O}{8}\right) + 4000S.$$

For the purpose of illustration, assume ultimate analyses as follows:

Kind of Fuel.	Percentage of			
	Carbon.	Hydrogen.	Oxygen.	Ash.
Anthracite.	93	1	1	5
Semi-anthracite.	87	2	3	8
Semi-bituminous.	79	3	8	10
Bituminous.	65	5	18	12

From this assumption it must not be understood that the percentage of ash recedes from the anthracite in any regular progression.

Then the total heat of combustion of such coal per pound, as calculated by the above formula, would be:

Anthracite,

$$h = 14,600 \times 0.93 + 62,000 \left(0.01 - \frac{0.01}{8} \right) = 14,275 \text{ B.T.U.}$$

Semi-anthracite,

$$h = 14,600 \times 0.87 + 62,000 \left(0.02 - \frac{0.03}{8} \right) = 14,174 \text{ B.T.U.}$$

Semi-bituminous,

$$h = 14,600 \times 0.79 + 62,000 \left(0.03 - \frac{0.08}{8} \right) = 14,014 \text{ B.T.U.}$$

Bituminous,

$$h = 14,600 \times 0.65 + 62,000 \left(0.05 - \frac{0.18}{8} \right) = 13,985 \text{ B.T.U.}$$

Very frequently the analysis merely states the amounts of fixed carbon, volatile matter and ash. Such an analysis is called the "proximate analysis." The volatile matter may be taken as consisting of marsh-gas, or CH_4 , without producing a sensible error. The total heat of combustion may then be calculated as below.

The following is an average of 24 analyses of Pennsylvania anthracites, as stated by Briton:

Fixed carbon.	91.05%
Volatile matter (CH_4).....	3.45
Ash and moisture.	5.50

The volatile matter consists of (by weight) $12 \text{ C} + 4 \text{ H}$ or $\frac{3}{4} \text{ C} + \frac{1}{4} \text{ H}$; therefore,

$$\frac{1}{4} \times 14,600 \times 0.0345 \dots\dots\dots = 377.8$$

$$\frac{1}{4} \times 62,000 \times 0.0345 \dots\dots\dots = 534.7$$

The heat-units due to the combustion of

$$\text{carbon, } 14,600 \times 0.9105 \dots\dots\dots = 13,293.3$$

$$\text{Total H. U. per pound} \dots\dots\dots = 14,205.8$$

In the proximate analysis the percentage of moisture should always be stated by itself, and not be added to any quantity such as ash. The percentage of sulphur should also be stated, but not included in the 100 percentage, since the sulphur gives an indication of "clinkering."

When moisture is present, as in nearly every case, then the total heat of combustion available is found by subtracting from the above results the heat necessary to evaporate this moisture. This may be done by the following formula:

Heat-units required to evaporate moisture in one pound of coal
= Percentage of moisture divided by 100, multiplied by

$$\{(212 - T_a) + 966 + 0.48 (T_f - 212)\}.$$

In which T_a denotes temperature of air in boiler-room;

T_f " " of furnace-gases;

966 equals latent heat of evaporation of water;

0.48 equals specific heat of steam under constant pressure.

The ash, as reported in analyses, consists of silica, oxide of iron, potash, alumina, lime, magnesia, soda, barium, phosphorus in phosphates, sulphur in sulphates, etc.

Liquid Fuels. These consist of the mineral oils, and their use has become more extended in the past few years. No doubt they would have a still wider field if there were less difficulty in obtaining a regular and constant supply.

The greatest quantities of the petroleum oils are produced in the United States and in Russia. Other oil fuels are blast-furnace oil, shale oil, creosote, green and similar tar oils.

The petroleum oils have a composition approximately as follows: Carbon, 86%; hydrogen, 13%; and oxygen, 1%. The specific gravity varies from 0.80 to 0.94, so that a gallon of oil weighs between 6.6 and 7.6 pounds.

As the total heat of combustion of a pound of oil varies from about 19,000 to 22,000 heat-units, this fuel has a theoretical evaporative power of from 19.6 to 22.7 pounds of water per pound of oil.

The oil is fed from tanks, and blown into the fire-box or combustion-chamber by means of a nozzle. This blast induces a current of air which assists in the combustion, although an additional supply is allowed to enter at the bottom of the furnace.

No grate is required, although a grate is often employed, which is covered with loose fire-brick. These bricks radiate off the heat and help to warm the air entering from below. Sometimes a small coal fire is employed to light the oil-blast in case the flame should be extinguished. Little or no change being required in the ordinary form of boiler-setting, beyond the introduction of the injector-nozzles, renders the boiler fit for the use of coal at any time in case the oil-supply should be interrupted.

The flames from the nozzles may be introduced into the furnace in horizontal, diagonal or vertical directions, as may be required.

The injector-nozzles are made in a great variety of forms, but all operate on the same principle.

The blast may be made by the employment of compressed air or steam. The latter method is the more popular, since the steam can be taken direct from the boiler, while the air necessitates the use of a compressor. The first boiler can be fired by a donkey boiler using coal, or by one of the main battery being fitted to use coal until the steam-pressure be sufficient to turn on the oil-blast. The main object sought is to blow the oil into the combustion-chamber in the form of spray, technically termed "atomizing" or "pulverizing" the oil. This pulverizing permits of its rapid combustion, which resembles the burning of a gas. Fig. 5 shows a fuel-oil burner operated by a steam-jet, and Fig. 6 one operated by an air-blast.

The direction of the blast should be such as to prevent the flame from impinging directly on the furnace-plates, and for this reason it is often blown against the pile of fire-brick mentioned above, sometimes called a "target."

The intensity of the flames is easily controlled by regulating the blast, which in turn controls the supply of oil.

Nearly all the pulverizing injector-nozzles are designed according to one of the following methods: The blast may enter at the centre and the oil on the outside, or the oil may enter at the

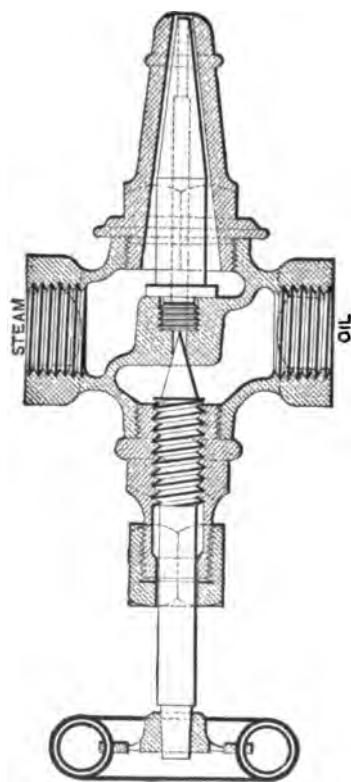


FIG. 5.—Rockwell Fuel-oil Burner, operated by steam reduced to between 40 and 80 pounds.

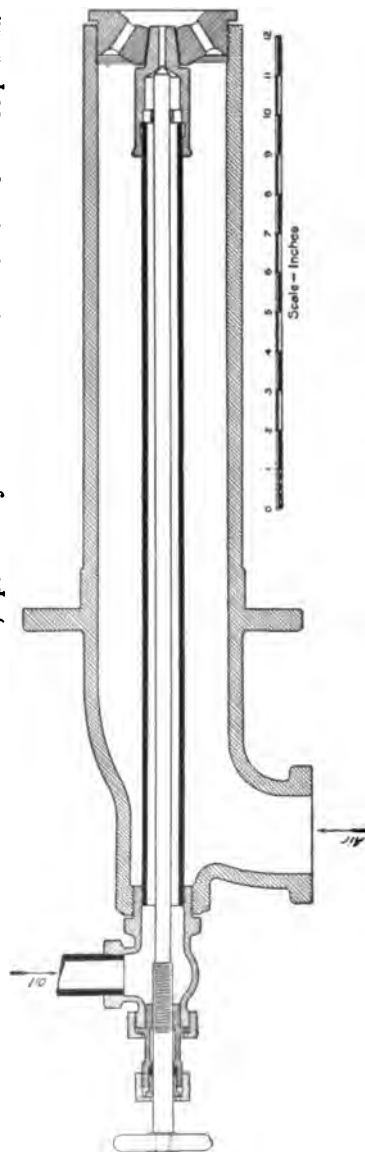


FIG. 6.—Lasse-Lovekin Fuel-oil Burner (Patented), operated by air at 1½ pounds pressure and with oil (heated to 175° F.) pumped to the atomizer at 15 pounds pressure.

centre and the blast on the outside. Practice has demonstrated that there is little if any difference in efficiency between the inside and outside methods. Most engineers, however, favor the blast being on the outside with the oil at the centre, since the blast in this position has the advantage of inducing a stronger current of air; and further, this method permits of the use of a circular opening for the oil, which is less liable to clog or choke up than the annular one, which would have to be used if the blast were at the centre.

Reference is made to a paper read by Col. Soliani entitled "Liquid Fuel for Marine Purposes," published in the Transactions of the Marine Congress, Chicago, 1893. It would appear from his experiments that the heating capacity of the crude petroleum oils is from 1.44 to 1.60 times that of average good coal, even after deducting the steam used to operate the pulverizers, which steam amounts to about 4 per cent of the total evaporation of the boilers. With the best forms of apparatus, Engineer-in-Chief George W. Melville, U.S.N., reports that this amount can be reduced to 2 per cent.

The boilers at the Chicago World's Fair, 1893, were fired with crude Ohio oil (petroleum), and the result, being the average of the daily reports, was:

Consumption of oil per hour.....	22,792 pounds
Water evaporated from 212° F. into steam at 125 pounds, per pound of oil.....	14.25 pounds
Equivalent evaporation from and at 212° F.....	14.88 pounds
Cost of oil per hour.....	\$56.20
Cost of oil per boiler horse-power per hour.....	0.0057
Cost of labor per boiler horse-power per hour.....	0.0006
Cost of boiler horse-power per hour.....	0.0063

Experiments were made by Mr. Holden on four locomotives of the Great Eastern Railway in England (Engineer, 15 February, 1895). The engines were express locomotives of the same type, and the experiments lasted for eight weeks, 1 December, 1894, to 12 January, 1895. Two of the engines burned coal, and two coal and oil, Fig. 7. The oil used was 'astatki,' or petroleum refuse.

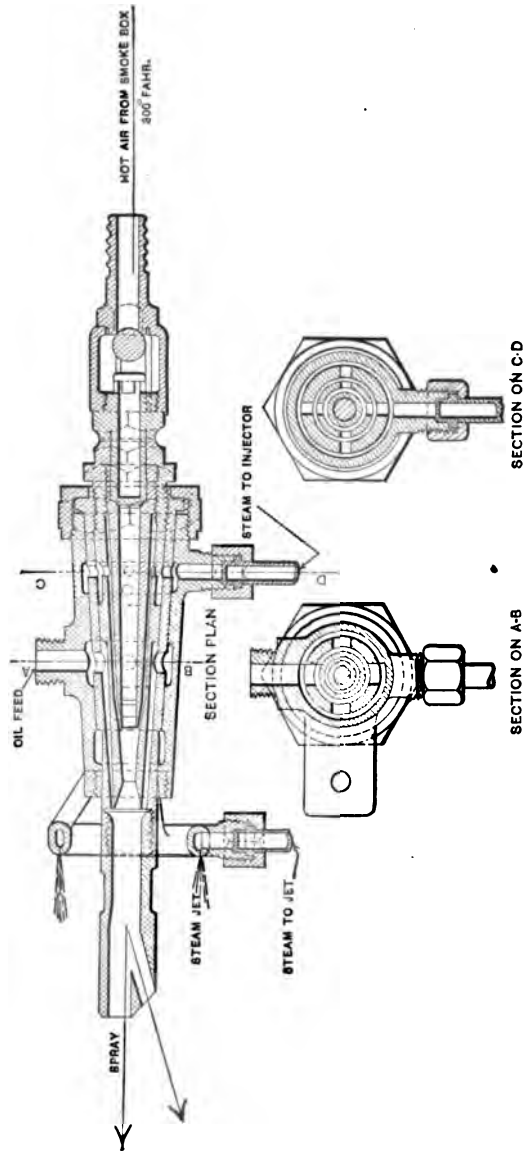


Fig. 7.—Steam Spray Atomizer for Fuel-oil—Holden System.

The results were:

1	pound of oil (maximum value)	was equivalent to	2.4	pounds coal
1	" " " (minimum value)	" " "	2.0	" "
1	" " " (average value)	" " "	2.18	" "

"In working * on Mr. Holden's system a thin coal fire is kept on the grate, and to assist in keeping the grate properly covered with a very thin fire lumps of chalk are placed on the grate when starting work for the day. The ash-pan dampers are kept very nearly closed, nearly the whole of the air required for supporting combustion entering either the injector-tubes or at the first door, which is kept open and fitted with an internal deflector just as when coal alone is being burnt. It is found that when burning the liquid fuel an exceedingly soft blast is required, and the blast-nozzle has to be materially larger than usual. The first experiments of Mr. Holden on liquid fuel were made at the Stratford shops of the Great Eastern Railway, using the by-products from the Pintsch's oil-gas works. The arrangement was next applied to three boilers of the locomotive type at Stratford, and on these its performances have been very satisfactory. The boilers are worked at 80 pounds pressure, and the comparative results of a week's working with coal only, and with coal and liquid fuel in combination, have been as follows: With coal (Staveley) only, the consumption for 63½ hours' work, including lighting up, was 156 cwt., or 275.1 pounds per hour. With the coal and oil in combination there were used in 60½ hours' work (including lighting up) 55 cwt. Staveley coal and 546 gallons of green oil, or an average of 101.8 pounds of coal and 9 gallons of oil per hour. With coal only, the evaporation was at the rate of 7.16 pounds of water per pound of coal, while with the coal and oil it was 8.91 pounds per pound of the combined fuel.

"With the liquid fuel it is found that the steam is kept up more easily and steadily than with coal alone, while the liquid fuel gives especial facilities for getting up steam rapidly if required.

"Various kinds of liquid fuel have been used, and the apparatus appears capable of dealing with any of the ordinary marketable qualities."

Some of the oil-burning locomotives of the Southern Pacific Railway are being equipped with the Heintzelman and Camp

* "Liquid Fuel in Locomotives," Engineering, 19 Oct., 1888.

arrangement (Fig. 8). The oil is supplied from Southern California, and is carried in a tank on the tender, from which the oil can flow by gravity to the atomizer. The novel feature is the placing of the burner at the front end of the fire-box, so that the flame is projected backward. The air-supply enters at the bottom near the back, and the products of combustion have to pass completely around the fire-box before entering the tubes. This arrangement favors complete combustion and high temperature, and appears to work most satisfactorily. It is found beneficial to warm the oil in a heater before it reaches the burner. A steam connection also is arranged to heat the oil in the storage tank, so as to warrant the flow of a thick oil by gravity to the atomizer.

A coal fire is not required on the grate of a modern oil-burning furnace.

Crude oil was used as a fuel at the San Francisco Midwinter Fair, 1894-95. (Engineering, 1 March, 1895.) The burner was a central tube about $\frac{1}{8}$ of an inch in diameter, through which the oil passed, and the flow was controlled by a valve. The steam-jet surrounded this oil-jet, and was given a rotary motion by means of guides, thus more completely atomizing the oil.

It was found that a number of small jets were more economical than one large one, because (1) the oil was more perfectly atomized, (2) the flame better distributed, and (3) better results could be obtained, when the boiler was not being worked at full capacity, by extinguishing some of the jets and using the others at full capacity, in place of throttling all the jets.

The best result was an evaporation of 15.13 pounds of water from and at 212° F. per pound of oil, while the average was 14.3 pounds. The maximum theoretical evaporative power of the oil was 20.7 pounds of water, equivalent to a heating power of 19,990 heat-units per pound.

Mr. R. Wallis, in a paper (Transactions of the American Society of Naval Engineers, Vol. IX, p. 781) read before the Northeast Coast Institution of Engineers and Shipbuilders, England, 1897, states that the air-blast, if heated, gives good results, but that more air than steam is required and that they are more noisy; that the air-blast gives a shorter flame and a more intense heat for a shorter distance from the flame. That the danger of an explosion of oil-gas in the combustion-chamber when lighting up, especially if the blast has been stopped for a short time only, is very much greater with air

than with steam. That there appears to be a better economy with steam-blasts, even including the water lost in the atomizer. That there is less liability of a breakdown with steam, since the air system is complicated by the compressor. That the greatest danger of explosion of oil-gas and consequent backflash from the furnace doors is in the relighting after the flames have been extinguished but a short time. That any small leakage of oil finding its way into the heated furnace gasifies and forms an explosive mixture with the air, and, if the lighting-up torch be introduced under these conditions, an explosion may result, with possible injury to the person holding the torch. That before lighting a furnace it should be well blown through with steam. That the steam-jets should be opened first, then the torch inserted, and finally the oil turned on. That there is no risk, even with the use of oil fuels having low flash-points, when these precautions are taken. That the average of a number of experiments, using the Rusden and Eeles' sprayer, gave the following heat-balance; the fuel being Russian astatki and the weight of steam required to spray one pound of oil being 0.3 lb.:

Heat-balance.	Heat-units.	Equivalent Evaporation from and at 212° F.
Total heat of combustion of 1 pound of oil:		
Carbon, $0.87 \times 14,500$	12,615	
Hydrogen, $0.12 \times 62,032$	7,444	
	20,059	20.7
Heat in waste gases at 450° F.:		
Carbonic acid gas. 3.19 pounds	269	
Nitrogen. 10.72 "	909	
Water vapor from combustion. . . 1.08 "	1,452	
" " " sprayer. 0.30 "	29	
Surplus air, taken at 20% 2.78 "	257	
	2,916	3.0
Heat lost by radiation, etc., by difference.	17,143	17.7
	1,687	1.7
Heat absorbed by water in boiler.	15,456	16.0

That, from a study of the annexed table, the heating value of the oil is about $1\frac{1}{2}$ times that of coal, but practice has repeatedly shown that oil fuel is equivalent in evaporative power to twice its weight of coal. That this difference can be accounted for to a great extent, as follows:

1. The combustion of the liquid fuel is complete, whereas that of coal is not, consequently in the former case there is no lost heat in smoke or soot.

2. There are no ashes or clinkers, and consequently no fires to clean with the accompanying loss of heat and drop in the steam-pressure.

3. The boiler-tubes are always free from soot and clean, and therefore always in the best condition for transmitting the heat from the gases passing through them to the water of the boiler.

TABLE VIII
COMPOSITION OF FUEL-OILS

Fuel.	Specific Gravity.	Chemical Composition.							Heating Power, B. T. U. per Pound.	Theoretical Evaporation.		Actual Evaporation in Pounds from and at 212° F.
		Carbon, Per Cent.	Hydrogen, Per Cent.	Nitrogen, Per Cent.	Sulphur, Per Cent.	Oxygen, Per Cent.	Ash, Per Cent.	Pounds from and at 212° F.		Pounds Air Required per Pound Fuel.		
Petroleum:												
Penn. heavy crude. . . .	0.886	84.9	13.7	1.4	20,736	21.48	14.56	16.0	
Caucasian light crude. . .	0.884	86.3	13.6	0.1	22,027	22.79	14.74		
" heavy crude. . . .	0.938	86.6	12.3	1.1	20,138	20.85	14.28		
" refuse. . . .	0.928	87.1	11.7	1.2	19,832	20.53	14.12		
Crude, avg. 15 samples	0.870	84.7	13.1	2.2	20,233	20.94	14.29		
Refined, average. . . .	0.760	72.6	27.4	27,531	28.50	17.93		
Scotch blast-furnace oil	0.920	83.6	10.6	0.1	9.4	18,590	19.20	17.93		
Coal:												
Avg. 98 samples, British.	1.279	80.4	5.2	1.2	1.25	7.87	4.0	13,968	14.46	11.34	8.13	

4. The temperature of the escaping gases may be considerably lower than is required to create the necessary draft for coal-firing.

5. The admission of air being under complete control, and the fuel being burned in fine particles in close contact with the oxygen of the air, only a small excess of air above that actually necessary for the complete combustion of the fuel is required. With coal, in order to insure as complete combustion as possible, a very much larger excess of air is required.*

* Passed Assistant Engineer John R. Edwards, U.S.N., delivered a lecture on "Liquid Fuel for Naval Purposes" before the Naval War College which contained much of value. (See Transactions American Society Naval Engineers, Vol. VII, p. 744.) The Pennsylvania Railroad made a number of experiments with oil fuels which were reported to be very satisfactory. The U. S. Navy Department has made a series of experiments and obtained very complete data, see Report of Liquid Fuel Board, U. S. Navy, published in Journal Am. Soc. Naval Engineers, August 1904.

The direction of the jet—horizontal, diagonal or vertical—does not appear to make any marked difference in efficiency. As a matter of convenience the horizontal direction seems best, since the nozzles can be made to pass through the ordinary fire-door opening or through the front casing, and necessitate no other change except that of the door. As the grate is undisturbed, a return to coal at any time is an easy matter.

A considerable economy is effected when the air-supply is heated before its mixture with the oil. This can be done by the escaping gases, the air being drawn through a pipe coil placed at the base of the flue. The air-supply pipe should be large, so as not to create loss by friction due to a high velocity through it.

The oil-tank should be located so that the fuel can readily flow to the nozzles, but should be lower than the burners to prevent accident from flooding. The oil can be drawn to the atomizers by the suction of the blast, but is generally pumped to positively control the supply and maintain a constant pressure. The oil can be burned with a natural or an artificial draft.*

There is no doubt that liquid fuel would be used to a much greater extent if the supply could be depended upon. In localities where it is cheap it is of great value as a fuel, but in most places its uncertain delivery and cost are prohibitive. For use in reheating and in heating furnaces for bending structural shapes and plates, as well as in annealing furnaces, its uniform heating power has created for it a marked value.

When compared with good coal the commercial efficiency of liquid fuel can be rated at 1 pound of oil to from 1.6 to 2 pounds of coal, which will include all the advantages due to the oil; so that at equal cost the oil can be preferred, due to its cleanliness. Good fuel-oils will evaporate from, say, 16 to 17 pounds of water from and at 212° F. per pound.

For the purpose of illustration assume:

Weight per gallon of fuel-oil, pounds.	6.8
Cost per barrel of 42 gallons delivered.	\$0.94

Then,

The cost of 2000 pounds of fuel-oil would be. . . .	\$6.58
---	--------

* For use of retarders with liquid fuel, see page 192.

Therefore, at a commercial efficiency of one to two, the values of the fuels are equal when the price of the coal delivered is \$3.29 per ton. This should include the cost of removal of ashes from the coal.

The advantages of liquid fuel are :

1. Reduction in number of firemen in proportion of 5 or 6 to 1.
2. Easy lighting of fires and more regular supply of heat.
3. The fires can be readily regulated to suit the demand for steam, and can be promptly extinguished.
4. The small proportion of refuse or ash and its easy disposal.
5. The storage-tanks can be located to best advantage, while coal-bins must be near the boilers.

The disadvantages may be stated as :

1. Danger from explosion and fire due to the vapors from the storage-tanks.
2. Loss due to evaporation.
3. The unpleasant odor.

Gaseous Fuels. These fuels have practically the same advantages and disadvantages as the liquid fuels, and like them afford a clean fire-room. In some special cases gas is purposely made for use as a fuel, but the general introduction of artificial gas for steam-generating is prohibited by its cost.

The waste gases from some metallurgical operations are used for heating steam-boilers. The gases are simply conveyed in a large pipe or flue, while at high temperature, beneath the battery of boilers, and there supplied with the requisite air to complete the combustion.

The natural gases are by far the most common of the gaseous fuels, and in the localities where found are used with great economy.

From whatever source, the gas is carried to the furnace in pipes, and ignited. The burner may be of any convenient shape, but usually is a plain tapered mouthpiece. Some of the best gas-burners are designed on the principle of the Bunsen burner, so as to insure more perfect combustion and a hotter flame. The flame may be horizontal, vertical or diagonal, to suit the situation, but it is best not to let it play directly against the furnace-sheets. As with oil, the grate may be covered with loose fire-bricks, which will greatly assist in warming the air-supply as it passes between them. If the grates are left in place, return can always be made to coal in cases of emergency. Sometimes a small coal fire is maintained on

the grate, so as to relight the gas should it become extinguished accidentally.

The gas in the supply-main is under a pressure varying from 1 to 8 oz., equivalent to $1\frac{1}{2}$ to 12 inches of water. When the gas-pressure exceeds 8 ounces it is usual to use a reducing-valve. A high pressure is apt to be wasteful as well as dangerous from explosions, and a reduced pressure is generally required by the fire insurance companies.

It has been found that one fireman can attend to boilers furnishing 200 H.P. with coal as a fuel; while with gas-firing one man can manage 1500 H.P., so that the reduction in labor is about $7\frac{1}{2}$ to 1 in large plants.

The heating powers of the gaseous fuels vary through wide limits. About 26,000 feet of natural gas or 100,000 feet of lean producer-gas are equivalent to one ton of good average coal.

The following tables, copied from Kent's "Mechanical Engineer's Pocket-book," are self-explanatory. Table IX may be considered as an 'average for the several gases, the figures being volumetric percentages; and Table X gives E. P. Reichhelm's experience, who states that under ordinary conditions in furnaces for drop-forging, annealing-ovens, and melting-furnaces for brass, copper, etc., the loss due to draft, radiation and the heating of space not occupied by the work is with gas of fair to good quality about 25 per cent.

TABLE IX

COMPOSITION OF FUEL-GASES

	Natural Gas.	Coal-gas.	Water-gas.	Producer-gas.	
				Anthracite.	Bituminous.
CO.....	0.50	6.0	45.0	27.0	27.0
H.....	2.18	46.0	45.0	12.0	12.0
CH ₄	92.60	40.0	2.0	1.2	2.5
C ₂ H ₄	0.31	4.0	0.4
CO ₂	0.26	0.5	4.0	2.5	2.5
N.....	3.61	1.5	2.0	57.0	56.2
O.....	0.34	0.5	0.5	0.3	0.3
Vapor.....	1.5	1.5
Weight in pounds of 1000 cu. ft.	45.60	32.0	45.6	65.6	65.6
Heat-units in 1000 cubic feet...	1,100,000	735,000	322,000	137,455	156,917

TABLE X

FUEL VALUES OF GASES

Kind of Gas.	No. of Heat-units in 1000 Cubic Feet Used.	No. of Heat-units in Furnaces after Deducting 25% Loss.	Average Cost per 1000 Cubic Feet.	Cost of 1,000,000 Heat-units Obtained in Furnaces.
Natural gas.....	1,000,000	750,000		
Coal-gas, 20 candle-power.....	675,000	506,250	\$1.25	\$2.46
Carburetted water-gas.....	646,000	484,500	1.00	2.06
Gasoline-gas, 20 candle-power.....	690,000	517,500	0.90	1.73
Water-gas from coke.....	313,000	234,750	0.40	1.70
Water-gas from bituminous coal....	377,000	282,750	0.45	1.59
Water-gas and producer-gas mixed ..	185,000	138,750	0.20	1.44
Producer-gas.....	150,000	112,500	0.15	1.33
Naphtha-gas, fuel 2½ gals. per 1000 ft.	306,365	229,774	0.15	0.65
Coal \$4 per ton, per 1,000,000 heat-units utilized.....				0.73
Crude petroleum, 3 cents per gallon, per 1,000,000 heat-units.....				0.73

CHAPTER IV

FURNACE TEMPERATURE AND EFFICIENCY OF BOILER

The Temperature. Color Test. Rankine's Method for Calculating. Dissipation of Heat Generated. Percentage of Heat Utilized. Results. Evaporation per Pound of Fuel and of Combustible. Practical Efficiencies.

The temperature obtained in a boiler-furnace depends on many conditions, including the design, fuel, moisture, amount of air supplied and rate of combustion.

The temperature can be measured by a pyrometer; by observing the melting or non-melting of substances, as specially made alloys; the increase in temperature of a given weight of water, when a block of metal of known weight is taken from the furnace and suddenly immersed; and from the color.

The color test must always be approximate, as so much depends on the eye of the observer and the darkness in which the bright object is viewed. The temperatures, as indicated by color, are usually thus stated:

Faint red	960° F.
Bright red	1300
Faint cherry	1500
Bright cherry	1600
Dull orange	2000
Bright orange	2200
White heat.	2400
Dazzling white.	2700 and over.

Professor Rankine's method of calculating the hypothetical temperature of the furnace is stated in the formula

$$W \times K_p \times T_f = h; \text{ therefore } T_f = \frac{h}{W \times K_p},$$

in which W denotes the weight of the products of combustion in pounds per pound of fuel;

K_p denotes the specific heat of the products of combustion under constant pressure;

h denotes the heat-units due to the combustion of one pound of the fuel; and

T_f denotes the temperature of the furnace in degrees Fahrenheit above that of the air.

The values of K under constant pressure are:

Carbonic acid	0.217	Air	0.237
Steam	0.480	Ashes, probably. . .	0.200
Nitrogen	0.244	Average value . . .	0.237

Assuming that a good average sample of anthracite coal contains 13,000 heat-units per pound, then the furnace temperature above that of the entering air would be:

With no excess of air

$$T_f = \frac{13,000}{13 \times 0.237} = 4219^\circ \text{ F.}$$

Note.—One pound of fuel plus 12 pounds of air equals 13 pounds of products of combustion.

With one-half quantity of air in excess

$$T_f = \frac{13,000}{19 \times 0.237} = 2886^\circ \text{ F.}$$

With whole quantity of air in excess

$$T_f = \frac{13,000}{25 \times 0.237} = 2194^\circ \text{ F.}$$

These temperatures should be corrected for moisture, when present, by deducting the heat required to evaporate it into steam. The assumption also is made that the specific heat remains constant, which may not be true for these high temperatures; and if it increases, the resulting temperatures will be correspondingly reduced. Such high temperatures as indicated above are not reached in practice, since the combustion is not instantaneous, is not all completed in the furnace as the flame and gases carry for some distance, and since the heat is being continually absorbed by the boiler.

The average temperature immediately over the fire varies from about 1400° F. to 1800° F. with natural drafts.

The quantity of heat generated in a boiler-furnace is dissipated in three ways: that utilized in the evaporation of water; that passing up the stack with the waste gases, thus supporting the draft; and that lost by radiation. From a well-designed boiler, properly set, the radiation loss is always small, being usually less than 5 per cent. The heat of the gases passing up the stack is necessarily lost for evaporation effects, but is essential to maintain the draft, unless an artificial draft be provided. The heat thus carried away in the gases must always be a considerable proportion of the total heat generated.

The percentage of heat utilized, that is, the efficiency of the furnace, may be determined thus:

Let T_f denote the temperature of the furnace;

" T_c " " " " " chimney-gases;

" T_a " " " " " air.

Neglecting the loss by radiation as being small, then approximately

$$\text{the heat utilized, in per cent,} = h_u = 100 \frac{T_f - T_c}{T_f - T_a}.$$

As an example, assume T_c to be 600° F.; T_a , 60° F.; and the values and conditions for T_f just given.

For no excess of air, $T_f = 4219 + 60 = 4279$:

$$h_u = 87.1\% \text{ of furnace heat.}$$

For one-half air in excess, $T_f = 2886 + 60 = 2946$:

$$h_u = 81.2\% \text{ of furnace heat.}$$

For whole quantity of air in excess, $T_f = 2194 + 60 = 2254$:

$$h_u = 75.3\% \text{ of furnace heat.}$$

These results clearly indicate that (1) the heat utilized decreases as the quantity of air admitted increases, but it is necessary to have some excess of air in order to burn perfectly all the carbon and hydrogen; (2) the heat utilized increases as the temperature of the chimney-gases decreases, and there should be sufficient heating surface to cool the gases as much as possible before they escape, although enough heat must be left to create a draft in

order to burn the fuel; and (3) the heat utilized increases as the temperature of the air admitted increases, so that it will be beneficial to heat the air before admission if it can be done without robbing the furnace of heat or the chimney of draft.

Economies of from 5 per cent to 15 per cent have been obtained by heating the air-supply before its admission to the furnace. When mechanical draft is used, this heating is often done by passing the air through conduits warmed by the escaping gases, the heat of which is not required in such cases for the maintenance of the draft. For the same reason, boilers show a slightly better rate of evaporation in summer than in winter. Some engineers have taken the air-supply from the engine-room, which utilizes part of the heat lost by radiation as well as assisting the ventilation.

If no losses occurred and all the heat were available for evaporation, then one pound of the best coal, containing, for example, 15,000 heat-units per pound, could evaporate from and at 212° F. (15,000 divided by 966) 15.5 pounds of water.

As in practice losses must exist, this result never can be obtained.

For sake of illustration, assume conditions to exist as before mentioned, with air-supply twice that theoretically required for complete combustion. Neglecting radiation losses, the available heat is then 75.3 per cent. The heat-units utilized per pound of coal burned would be $13,000 \times 0.753 = 9789$.*

Assume that the boiler is generating steam at 85.3 pounds by the gauge, and that the feed-water has a temperature of 100° F. Then the total heat of evaporation would be, in heat-units per pound:

From the Steam-table, $1181.8 - (100 - 32) = 1113.8$, or by Regnault's Formula, should a table not be at hand,

$$h_{2,1} = 1091.7 + 0.305 (327.6 - 32) - (100 - 32) = 1113.8.$$

The evaporation, then, of one pound of coal under these conditions would be:

$$\frac{\text{Heat-units per pound of coal}}{\text{Heat-units per pound of steam}} = \frac{9789}{1113.8} = 8.78 \text{ pounds of water.}$$

* Results calculated in this manner are greater than would occur in practice, as nearly always there is some moisture present. Furthermore, it is based on a constant specific heat, which may not be true for high temperatures as was mentioned above.

Multiply this result by the factor of evaporation to find the equivalent evaporation from and at 212°. The factor for the case is 1.153.

Equivalent evaporation from and at 212° = $8.78 \times 1.153 = 10.12$ pounds.

The following are the result of three tests, the first from Transactions Am. Soc. M. E. 1891, page 990, and the others from the author's note-book:

1. Type of boiler. Return tubular
 Average steam-pressure. 90 pounds
 Feed-water temperature. 147° to 150° F.
 Evaporation from and at 212°, with
 pea coal. 9.9 pounds
 Evaporation from and at 212°, with
 anthracite lumps. 10.2 pounds
2. Type of boiler. Return tubular
 Average steam-pressure. 34.7 pounds
 Feed-water temperature. 52.5° F.
 Coal. Cross Creek anthracite
 Evaporation from and at 212°. 7.8 pounds
3. Type of boiler. Return tubular with superheater
 Evaporation from and at 212°. 8.5, 9.4, 8.9 pounds under varying conditions

Locomotives evaporate, when in clean condition, about 6 to 7 pounds of water per pound of coal.

The rate of evaporation is frequently expressed "per pound of combustible" in place of "per pound of fuel."

"**By fuel**" is meant the fuel as fed into the furnace, and, so to speak, the term is used in the gross sense.

"**By combustible**" is meant the net fuel, or that which is consumed on the grate. It is the difference between the weight of fuel as fed into the furnace and the weight of the refuse as removed. This refuse consists of ashes, fuel that falls into ash-pit through grate-bars, and the dust or soot that may pass through the boiler into the stack. Owing to the difficulty of weighing the latter, it is seldom considered except in a few very elaborate boiler tests.

The efficiency of the boiler, that is, the percentage of the total heat of combustion which is utilized for evaporation, varies considerably.* Some boilers show an efficiency of less than 50 per cent,

* George H. Barrus' work entitled "Boiler Tests" will be found valuable

but such low results are rather exceptional, being traceable to smallness of size or to poor design or setting. Under ordinary conditions of practice, efficiencies from 50 per cent to 70 per cent may be considered as poor to fair; from 70 per cent to 75 per cent as good; and over 75 per cent as excellent.

The efficiency of a boiler can be determined for any condition of operation, by measuring the water evaporated and the fuel burned during the same time.

Then the efficiency in per cent is;

$$\frac{\text{Heat absorbed by the water}}{\text{Heat of combustion of the dry fuel}} \times 100 =$$

$$\frac{\text{Total heat of evaporation} \times \text{rate of evaporation}}{\text{Heat of combustion of 1 pound of dry fuel}} \times 100$$

The above result is really the efficiency of the boiler and grate.

The efficiency of the boiler, without the effect of the grate, in per cent is:

$$\frac{\text{Heat absorbed by the water}}{\text{Heat of combustion of the combustible}} \times 100 =$$

$$\frac{\text{Total heat of evaporation} \times \text{rate of evaporation}}{\text{per pound of combustible}} \times 100.$$

$$\frac{\text{Heat of combustion of 1 pound of combustible}}{\text{Heat of combustion of 1 pound of combustible}} \times 100.$$

This latter result omits from consideration the fuel lost through the grate in an unburned condition, and should be the one used as the standard of comparison for boiler trials. The word "combustible" is here used to mean the fuel without moisture and ash.

for reference in this particular, as it contains records of tests on many different types under variable conditions

CHAPTER V

BOILERS AND STEAM-GENERATORS

General Conditions. Classification. Horse-power. Centennial Standard. Am. Soc. M. E. Standard. Heating Surface. Ratio of Heating to Grate Surface. Evaporation per Square Foot of Heating Surface. Design. Description of Certain Boilers. Proportioning a Boiler to Perform a Given Duty. Steam-space. Priming. Water Surface.

Steam-boilers and Steam-generators are essentially metallic vessels in which water is heated and converted into steam. The term "boiler" is generic, but when used in its restricted sense it refers to those boilers which are more properly "metallic vessels" in which there is a considerable mass of water in relation to the capacity. On the other hand, "steam-generator" is the term made applicable to that class in which the mass of water is relatively small to the capacity, and confined principally by tubes and parts of small dimensions.

Since the cost of fuel, no matter what kind may be used, forms so great a proportion of the total cost of the output or product of every plant, it is all-important that the boiler be economical and efficient. The greatest care should be given to the design or the selection of a boiler for each particular case. The fuel should be burned to best advantage, so as to generate the greatest furnace temperature; and the boiler should be so arranged as to abstract this heat from the products of combustion, permitting no more to escape than may be necessary for the maintenance of the draft. The gases and air should be thoroughly mixed by means of bridge walls or other devices; and generous proportions should be given to the combustion-chamber in order that the combustion may be completed before the gases become cooled by contact with the boiler surfaces. In order to absorb the heat of the products of combustion there should be plenty of heating surface, the arrangement of which should be such as to encourage the proper circulation of the

water in the boiler, and at the same time to prevent the gases from making a short passage to the stack. While making the provision for the gases to reach every portion of the heating surface, care must be exercised lest too great a resistance to draft be created. The surfaces from which radiation may occur should be well lagged or clothed.

The history of the steam-boiler shows that the gradual development has been in the direction of increase of heating surface, reduction in weight and higher pressures. Future improvement no doubt will follow these tendencies.

It can be frankly stated that there is no "best" type of boiler, although one kind may be much better suited than another for some particular class of work, or for operation under certain fixed conditions.

Boilers are usually classified as being either

Externally fired or

Internally fired.

The distinctive feature is whether the fire on the grate is external to the boiler proper, as, for example, the plain cylindrical boiler or the horizontal return-tubular boiler; or whether the fire is internal, as in the Scotch boiler or in the Lancashire boiler. There are some types which are more or less difficult to place under either head, but which belong in part to both.

Another classification is

Fire Tubular or

Water Tubular.

The distinctive feature under these headings is the use of the tubes or flues. If the hot gases pass through them, the water being on the outside, the boiler is said to be of the fire-tubular type. If the water be in the tube with the hot gases on the outside, then the boiler is water-tubular.

As before, these names do not cover all types for some boilers, as the plain cylindrical, have no tubes; and again there are boilers, known as "compound" boilers, which are a combination of the two classes.

Horse-power of Boilers. For the sake of convenience it has become necessary to adopt a rating for boilers, and custom has adopted the term "horse-power" to designate the unit of compari-

son. Strictly speaking a horse-power is the unit of rate of work, and its application to the boiler is a misuse of the expression, since a boiler does not perform work in the sense of "overcoming of resistance through space." The term has become so general, however, that it must be retained, and if clearly understood is as good as any other. It is to be urged that when used the term should always be "boiler horse-power," as then there is little likelihood of its being misinterpreted. The horse-power of an engine has no relation whatever to the boiler horse-power of the boilers furnishing the steam.

The builders of steam-boilers usually rate their boilers at one boiler horse-power to every ten, twelve and one-half or fifteen square feet of heating surface. Such ratings give little idea of the boiler, as so much depends on arrangement of surface, rate of combustion and the like. While ten to fifteen square feet of heating surface may correspond to the average engine horse-power, it must not be forgotten that an engine horse-power is often developed on three square feet and even less. A far better plan is to rate the boiler according to the quantity of water that it will evaporate.

The "Centennial" standard, being that used by the Committee at the Exposition in Philadelphia, 1876, was an assumption of 30 pounds of water evaporated per hour from feed-water at 100° F. into dry steam at 70 pounds pressure by the gauge, as being one boiler horse-power.

The American Society of Mechanical Engineers' standard is practically the same, being $34\frac{1}{2}$ pounds of water evaporated per hour from a feed-water temperature of 212° F. into dry steam at the same temperature. The evaporation being dependent on the draft, the Committee of the Society recommended that a boiler rated at any horse-power should develop that power when using the best coal ordinarily sold in the market where the boiler is located, fired by an ordinary fireman without forcing the fires, while exhibiting good economy; and, further, that the boiler should develop at least one-third more than its rated power when using the same fuel and operated by the same fireman, the full draft being employed and the fires being crowded; the available draft at the damper, unless otherwise understood, being not less than $\frac{1}{2}$ inch water column.

Heating Surface. Boiler-makers generally measure up the heating surface on the outside of all tubes and flues, as the result is

greater than when the inside areas are considered. It is a mooted question which is the better method to pursue. Many argue that the side next to the fire should be considered, because it is technically the correct heating surface. Such a system would result in outside area for water-tubular, and inside for fire-tubular boilers. Others say that the outside should be considered in all cases, because (1) it is simpler, since the diameters of all boiler-tubes are catalogued and ordered on outside measurement; (2) there is no need of being so very accurate, since the heat-transmitting power is not constant, but varies with position and thickness; and (3) more is gained for comparison by adopting a uniform method for all cases. Neither method covers every form. For instance, what should be considered the heating surface of such special shapes as the *Serve* tube (Fig. 55)? Its heating surface is certainly greater than the outside area, but is not effectively equal to the inside area when the surface of the ribs is included. It may be asked what is its heat-transmitting surface when partly filled with soot between the lower ribs?

The American Society of Mechanical Engineers favors a computation of area of surface of shells, tubes, furnaces and fire-boxes in contact with the fire or hot gases. The outside diameter of water-tubes and the inside diameter of fire-tubes are to be used in the computation. It is best to compute the total area as accurately as possible, and state separately the square feet of effective water-heating surface and of steam or superheating surface. All surfaces below the water-line having fire or hot gases on one side and water on the other are water-heating surfaces; and all above the water-line having hot gases on one side and steam on the other are superheating surfaces. Any surface below the line of the grate to which the flames do not have access should not be considered, nor any surface that may be covered by brickwork or bridge walls. Only three-quarters of the bottom of horizontal-shell boilers set in brickwork tangent to the shell, should be taken, as the part on each side next to the brick walls is not effective. If the brick walls are corbeled off from the shell, then the full area may be taken. For corrugated and Morison suspension furnaces, use the area due to the mean diameter and add $14\frac{1}{2}$ and $9\frac{1}{6}$ per cent for additional surface respectively. For the Purves ribbed furnace use the outside diameter of flat part and add 11 per cent.

Since the tube-sheets are not very effective, many engineers

neglect them altogether and figure the tube surface on the extreme length of tube. While the result between this method and that of computing the area of tube-sheets between tubes plus the area of tubes figured on their length between tube-sheets is nearly the same, the author favors the latter method as being the more uniform and accurate for all cases. When the feed-water is heated by the products of combustion, such area of surface should be stated, but not included as water-heating surface.

When the heating surface consists nearly all of tube surface, the efficiency of the tubes as measured by the total evaporation will be found to vary with their length, and nearly in the following ratio:

Length of tubes, in diameters..	60	50	40	30	20
Relative water evaporation....	1.00	0.91	0.83	0.75	0.67

If the length exceeds 60 diameters, the evaporating efficiency of the increased heating surface falls rapidly. It is advisable, therefore, not to make the tubes over 50 or 60 diameters in length, and to use more tubes or tubes of a different size in order to make up any deficiency in surface.

The text-books usually state the ratio of heating surface divided by grate surface for various boilers to be:

For plain cylindrical.....	from 10 to 15
“ Cornish.....	“ 20 “ 25
“ Lancashire.....	“ 20 “ 30
“ Cylindrical tubular.....	“ 25 “ 40
“ Marine, natural draft.....	“ 28 “ 35
“ “ forced draft.....	“ 35 “ 50
“ Water-tubular.....	“ 30 “ 40
“ Locomotive, with blast.....	“ 50 “ 130

These figures can be taken as representing average results, but it is evident that for any particular case the ratio must depend on the kind of fuel and on the rate of combustion. The dominating idea is to absorb all the heat possible, leaving only sufficient for purposes of draft.

The quantity of water that will be evaporated from each square foot of heating surface is a very variable quantity, and can only be foretold, even approximately, by those having had considerable experience. As a guide, however, it may be stated that each square

foot of heating surface may be expected to evaporate per hour, under conditions usually obtained in practice:

In stationary boilers, moderate draft.	from 2 to 3 pounds
" " " fairly strong draft.	" 3 " 4 "
" " " strong draft.	" 4 " 5½ "
" " " very strong draft.	" 5½ " 8 "
" marine boilers, moderate draft.	" 3 " 5 "
" " " strong draft.	" 5 " 7 "
" " " very strong draft.	" 7 " 8 "

Such results are obtained by dividing the total evaporation by the water-heating surface. All the heating surface is not, however, equally effective, but some portions evaporate many times the quantity of others. Surfaces which are horizontal or nearly so, and those so placed that the gases will not pass along them in "stream" lines, but rather impinge against them, are considered the best. Also, those parts which are exposed to a convection current washing away bubbles of steam as fast as formed, are better than those against which the water is more nearly quiescent.

The water-heating surface varies in efficiency according to character and location. Shell heating surface is more efficient than the ordinary tube heating surface. Thus 9 square feet of plain cylindrical boiler surface is about equivalent to a B.H.P., while 12 to 15 square feet are required in horizontal return-tubular boilers. The former type of boiler, however, fails in comparison on a basis of economy in steam production.

George H. Barrus states in "Boiler Tests" that little or nothing is gained in evaporation with anthracite coal when the ratio of heating to grate surface is greater than 36 to 1, and considers that about the best ratio for horizontal tubular boilers. The result was based on average results from a number of tests, all having low temperatures for the escaping gases and rates of combustion exceeding 9 pounds with natural draft, but not exceeding 12 pounds. When the ratio was increased to 65 to 1 there was an actual loss. With bituminous coal in horizontal tubular boilers he found better economy with greater ratios; thus at 42 to 1 the gases still retained high temperatures, and the best result in his tests was at a ratio of 53 to 1. He also found that in the same boiler the temperature of escaping gases was always higher with bituminous than with anthra-

cite coal, showing that more heating surface is needed for the former. Therefore for bituminous coal the ratio should be between 45 and 50 to 1, when rate of combustion is about 10 to 12 pounds of coal per square foot of grate per hour. He further states that these proportions gave good results in vertical boilers; and sees no reason to change them for the sectional or water-tubular class.

From a careful study of the subject it appears that there is no economy in forcing boilers beyond their proper capacity, as then the loss of heat up the stack becomes excessive from lack of heating surfaces in relation to the coal burned. In general, boilers are too often operated with a deficiency of heating surface. The amount of this heating surface should increase with the rate of combustion, starting with the values mentioned by Barrus. In the author's practice the question arose how to diminish the fuel account in a certain plant. No test or data could be obtained beyond simple inspection with the eye. It was recommended to add 50 per cent to the battery of boilers, everything else being retained as it existed. The change caused a remarkable saving, due to a better ratio of heating to grate surface and a consequent reduction in loss due to high temperatures of escaping gases, as well as to a saving of coal raked through the grate-bars by the continual forcing of fires.

The Design. While the design of a boiler is not a difficult problem, it is necessary in order to attain success to keep in mind the fundamental principles involved. The engineer should have some practical knowledge of the boiler-maker's methods of handling the pieces, or the finished boiler will differ from the drawing in many details. This knowledge can only be obtained by practice, and it is this extended experience which should enable the engineer to produce a better design than the manufacturer, since he has become familiar with the methods of many boiler-shops and is thus enabled to select the best details, while the boiler-maker is apt to favor the methods of his own practice to which usage has made him accustomed.

The engineer should have clearly in mind just what results he wishes to obtain before he commences his plans. The points to be determined may be stated thus:

First. The rate of combustion desirable or practicable. This rate depends upon the draft.

Second. The type of boiler. This will depend largely on the

quality of feed-water, liability for overwork, cost of manufacture, erection, operation and maintenance.

Third. The quantity of steam demanded. Upon this will depend the amount of power to be put into one boiler, and therefore the actual number required.

Fourth. The kind of fuel and the quantity, as well as the efficiency that may be expected.

Fifth. The ratio of heating to grate surface and the actual areas required to burn the fuel and absorb the heat.

Sixth. The nature of the setting and the manner of making the steam and water connections.

Seventh. The necessary drawings, showing arrangements of all details and connections. These drawings are usually made to a scale of one inch to one foot, with details to larger scale as may be convenient.

In making the design, care must be taken to give the maker ample room for laps, flanging, etc., otherwise the dimensions will not be as expected. This is a very common fault with good designers, and is especially liable to occur when only outline plans are made. Since most boiler-makers lay out the work to full scale commencing at the bottom, all errors are cumulative, with the result of raised water-line and diminished steam-room. The latter is seldom made too large, so that the fault is serious and liable to produce foaming.

The scantlings will be treated in another chapter. The principles involved are simple, and although attempts are made to obtain improved results by changes in details and in general arrangement, little if any gain is produced by complication. The best boilers of the various well-known types, although of diverse forms, are practically equal when tested under conditions suited to the designs, the results published in trade catalogues notwithstanding.

In preparing the design, it is well to keep in mind the following suggestions, remembering that any plan must be one of "give and take." Therefore weigh the advantages against the disadvantages, and settle each question as it occurs.

First. Have as few joints and double thicknesses of metal exposed to the fire as possible.

Second. Protect furnace, flues and tubes from sudden impingement of cold air.

Third. Provide against delivery of cold feed-water upon hot plates or tubes.

Fourth. Provide for a good circulation in order to carry away steam from the heating surfaces as soon as formed.

Fifth. Provide ample passages for upward as well as for downward currents, and arrange that they shall not interfere.

Sixth. If of sectional type, provide ample passages between the various sections, so as to equalize pressure and water-level in all.

Seventh. Provide sufficient water surface to allow the steam to separate quietly.

Eighth. Provide ample steam-room, so as to maintain a constant water-level and to prevent foaming.

Ninth. Provide for carrying the hot gases to all parts of the heating surfaces and prevent them from making a short circuit to the chimney, or while still at high temperature from reaching tubes containing steam only, unless such surfaces are arranged for super-heating.

Tenth. To so arrange the parts that they may be constructed without mechanical difficulty or excessive expense.

Eleventh. To select a form that will not unduly suffer under the action of the hot gases.

Twelfth. To make all parts accessible for cleaning and repair.

Thirteenth. To give each part, as nearly as possible, equal strength. Such parts as are most liable to loss from corrosion, etc., should be made the strongest, so that the boiler when old shall not be rendered useless by local defects.

Fourteenth. To adopt a reasonably high factor of safety in proportioning the parts.

Fifteenth. To endeavor to secure careful, intelligent and efficient management.

The different types and arrangement of steam-boilers are so numerous as to be impossible to describe all. A few of the more general types will be selected as illustrations.

The Plain Cylindrical Boiler consists of a long cylindrical shell built up with plates (Fig. 9). The heads are nearly always bumped, so that the boiler is self-sustaining and requires no stays.

The shell is set in brickwork, and is supported by rods from above. The eye in the rod end engages a hook riveted to shell, or the rod end may have a hook formed on it and catch a ring riveted

to the top of shell. The rods can be made with turnbuckles or with a thread and nut on upper end so as to adjust the length.

The boilers should be slightly inclined to facilitate draining, one inch or two inches in the total length being generally sufficient.

The grate is usually made common to two or three boilers, a feature which permits the use of cheap grades of fuel. The objection is the necessity of laying off the other boilers in order to repair or clean one. The space between the boilers is closed by means of arched fire-brick resting on the shells

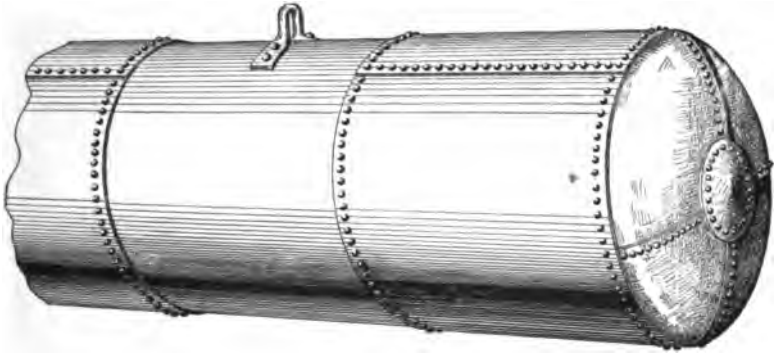


FIG. 9.—End of a Plain Cylindrical Boiler.

These boilers seldom exceed 42 inches in diameter by 50 feet in length. A common size is 36 inches in diameter by 30 feet long.

They are cheap, easily handled and quickly cleaned. As there is difficulty in obtaining heating surface, and as they are not economical except in first cost, this type is little used unless in connection with the waste gases obtained from certain metallurgical works. They require large floor-space, which is often an objectionable feature.

The Horizontal Return-tubular Boiler is a form in extensive use in American practice (Figs. 10, 11, and 12). It is simple, inexpensive, and, when properly handled, is durable. It contains considerable heating surface for the space required, and is economical.

The shell varies in thickness from $\frac{1}{4}$ inch upwards, according to pressure. The heads are usually made $\frac{3}{8}$ inch thick for boilers less than 36 inches diameter; $\frac{7}{8}$ inch, from 36 inches to 60 inches; $\frac{1}{2}$ inch, from 60 inches to 72 inches; and $\frac{9}{8}$ inch for all over 72 inches.

The fire is on an external grate, and the products pass beneath

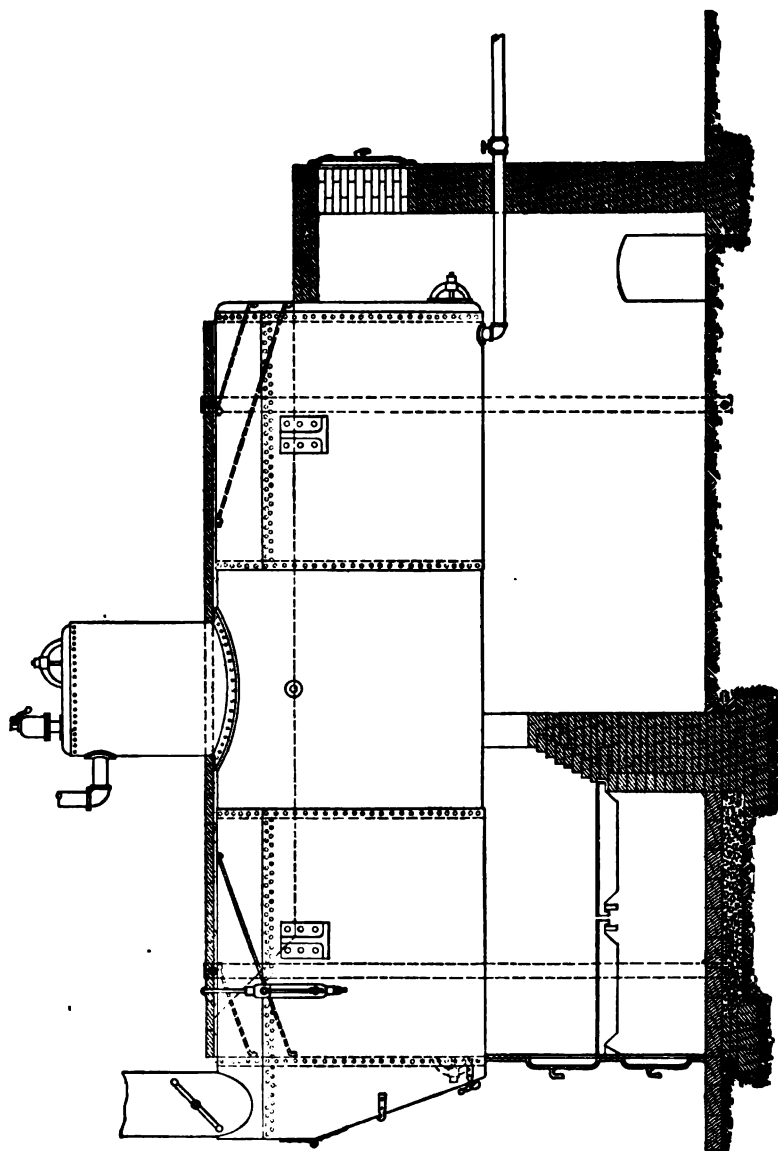


Fig. 10.—Horizontal Return-tubular Boiler, with extended or half-arch front.

the shell to the back end, and return through tubes to the front, where they pass into the smoke-connection. Occasionally the products are again made to pass along the top of the shell to the rear end, but this practice is rare, as it makes an inconvenient arrangement for reaching the steam and other attachments on top of the shell.

These boilers are made with either an extended (Fig. 10) or flush

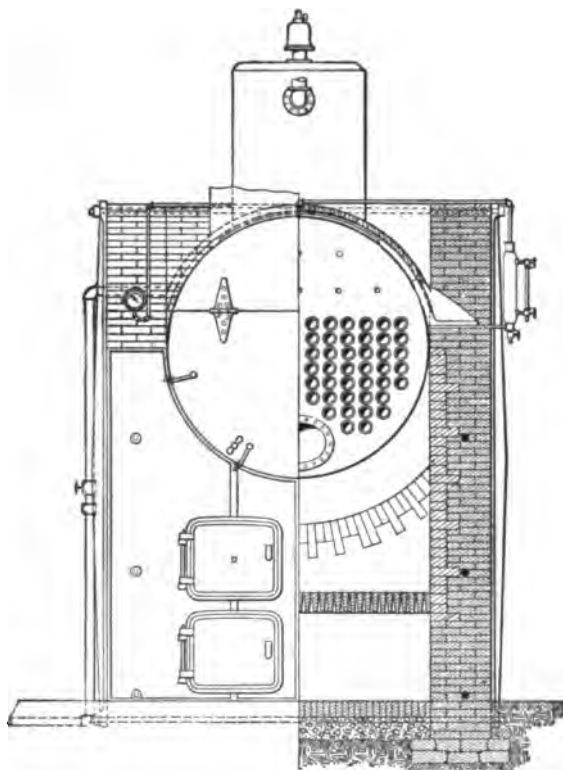


FIG. 10a.—Horizontal Return-tubular Boiler.
Front and section of Fig. 10.

front (Figs. 11 and 12). The former is the cheaper, and is made by extending the shell plating about 15 inches beyond the front head, so as to form a smoke-box, to the top of which the smoke-flue is connected by an angle or cast-iron ring. The front is closed by a hinged door, made high enough to expose all the tubes for cleaning.

With the flush front, the head end of the boiler can be the

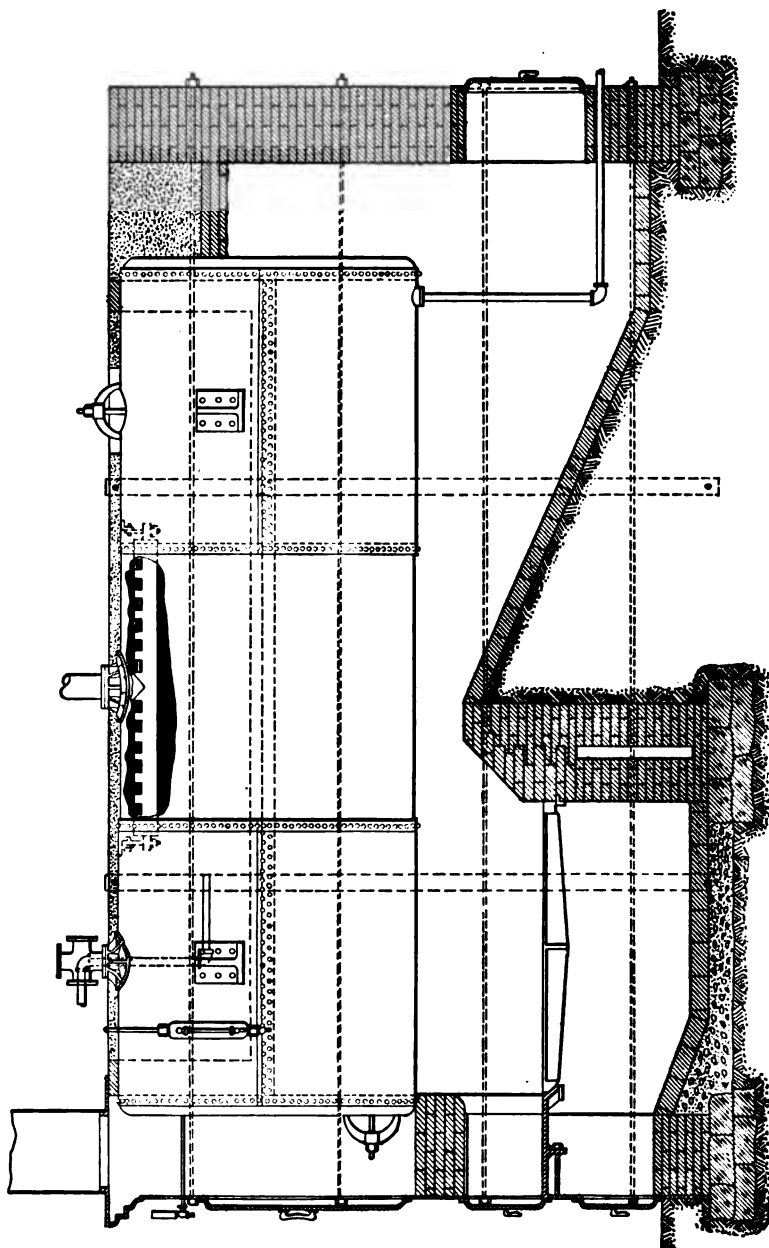


Fig. 11.—Horizontal Return-tubular Boiler, with flush or full front.

same as the back end, with the smoke-box constructed in the brick setting.

The fronts in either case are generally made of cast iron, with panel-work more or less elaborate. The half-front fits around the extension smoke-box, while the flush front extends to the top of the

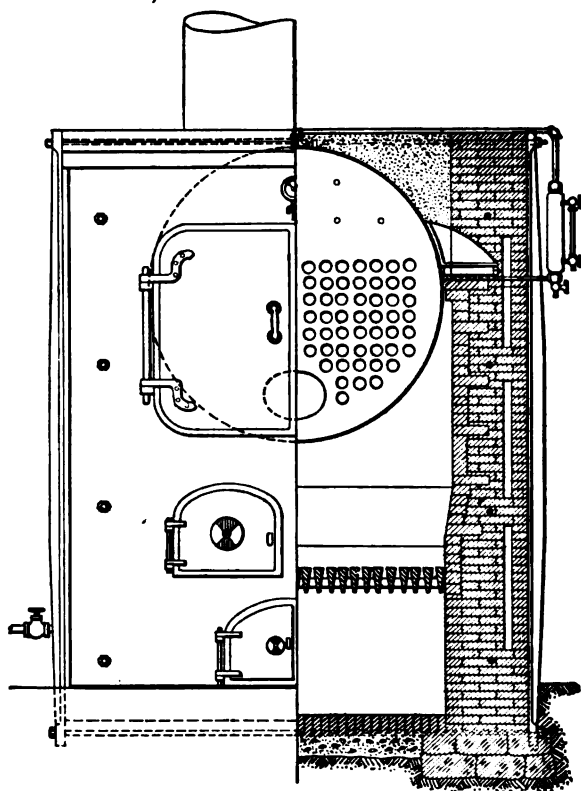


FIG. 11a.—Horizontal Return-tubular Boiler.
Front and section of Fig. 11.

brick setting. The flush front has a door, usually made in halves with hinges on the sides, to expose the tube-ends.

The settings are made of brick. The shell frequently rests on the side walls, being supported by cast-iron lugs, cast to fit the shell. There are two lugs on each side, although some very large boilers have three.* The lugs are generally one inch thick, with a stiffening

* For boilers 66 inches in diameter and larger, it is best to use four lugs on a side, placing them in pairs close together.

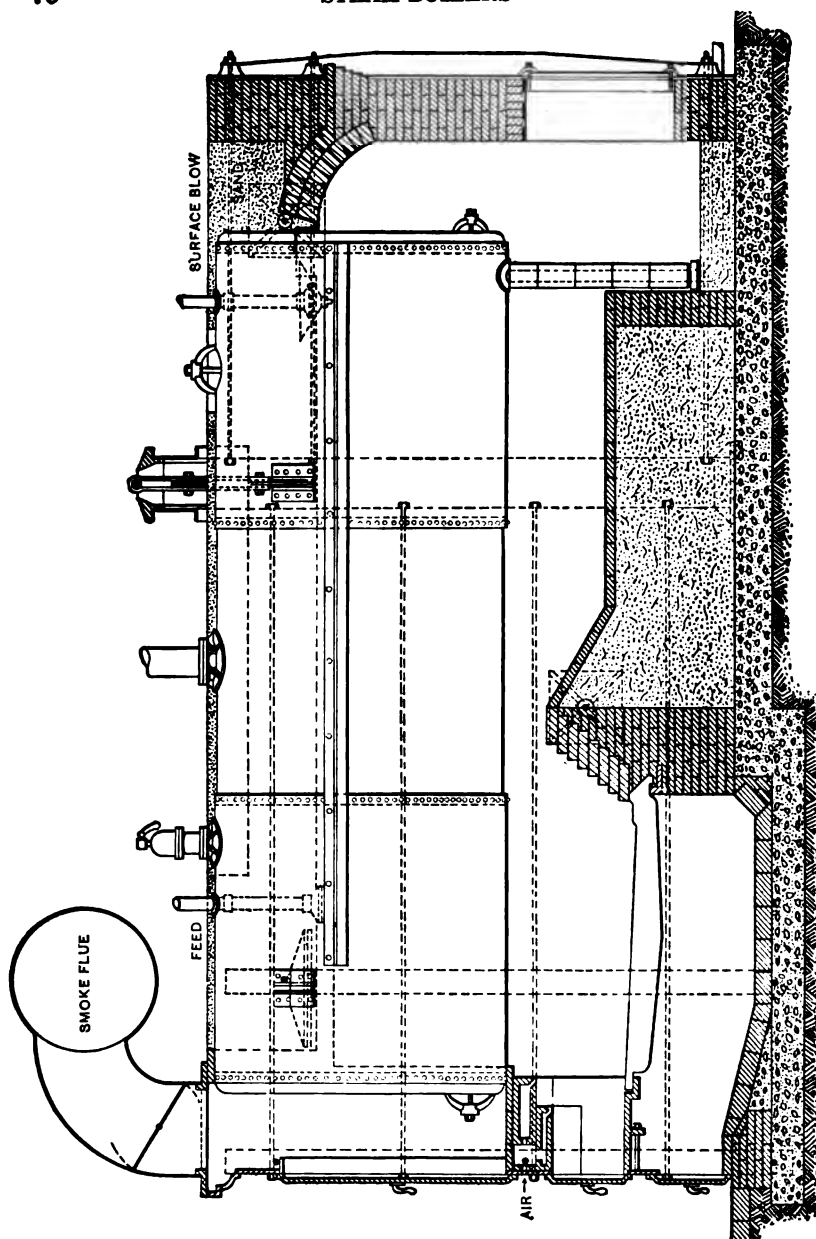


Fig. 12.—Horizontal Return-tubular Boiler, with a link suspension.

web $\frac{3}{4}$ to 1 inch thick. These lugs are bolted to the shell, although small lugs are sometimes riveted on. A bolt screwed through the shell and lug, fitted with a nut, makes a good arrangement. The bottom flange is from 8 to 14 inches long by 6 to 12 inches wide.

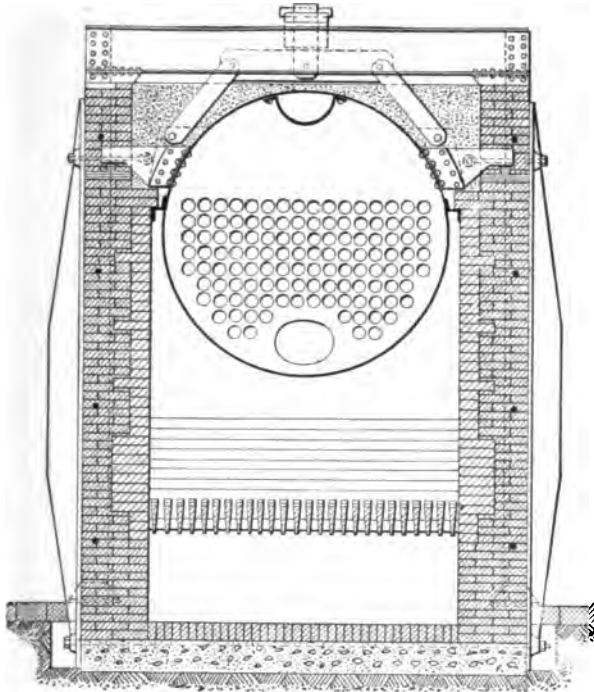


FIG. 12a.—Horizontal Return-tubular Boiler.
Section of Fig. 12.

The lug rests on a cast-iron or steel plate built in the brickwork. Cast iron is the better. The plate should have a surface of at least one square foot. When of steel it should be $\frac{3}{8}$ inch thick, and when of cast iron $\frac{3}{4}$ or 1 inch thick. The front lugs rest directly on the plates, but the back lugs should have rollers not less than one inch

in diameter, between lug and plate, to let the boiler accommodate itself by expansion and contraction.

This method is open to many objections. It is difficult to build up the brick walls under the boiler to just the proper height for the several lugs. If one is too high or too low, it will cause a twisting strain on the shell. Furthermore, it is almost impossible to get the rollers to bear true and even. One is apt to carry the weight and the others be loose. Also, the projecting rivet-heads are liable to injure the brick setting as the boiler expands or contracts.

These defects in setting can be avoided by hanging the boiler on suspension links. A method advocated by Mr. O. C. Woolson

(Trans. Am. Soc. Mechanical Engineers, Vol. XIX, 1898) consists in carrying the front brackets on the side walls and suspending the rear end from a link supported by channels placed across the top of the setting (Fig. 12). The front brackets are long enough to put the weight on the outside wall, so that the inside fire-brick lining can be taken down at any time for renewal. The buck-staves or binder-bars at the front are fastened to the supporting brackets, so that they move as the boiler expands and relieve the brickwork. The side channels at the rear, which carry the cross-channel, form the buck-staves for that end. This method is really a three-point suspension. The rivet-heads are kept away from the brick walls by using a 3-inch Z bar riveted to shell, against the smooth side of which the side walls are built.

These boilers should be set slightly inclined toward the end having the bottom blow-off, so as to drain freely. About one inch is sufficient.

The distance between the back head and the rear wall should be about 16 inches for boilers less than 58 inches diameter, and from 18 to 24 inches for larger ones.

The distance from top of grate to under side of shell should be as great as possible so as not to chill the gases. It is often made too small. Soft coals require a greater height than hard coals. The distance is usually 26 to 30 inches for grates 4 feet in length, and should be increased in proportion to length. For soft coals this distance should be increased by about 20 per cent.

The combustion-chamber behind the bridge wall may be filled up with earth or ashes, so as to keep the hot gases near the shell, but its actual shape appears to make little difference. It is well, however, to pave the floor of the combustion-chamber, that the heat may be radiated back as much as possible. A clean-out door, large enough for a man to pass, will be found convenient at the rear end of the combustion-chamber. This door should be carefully made and set, so as to be air-tight.

The side and rear walls are best made double, with an air-space between them about 2 or 4 inches wide. With a 2-inch space the total width of wall would be $18\frac{1}{2}$ inches. About every two feet headers should be run from wall to wall, but the walls should not be bonded together. The air-space prevents radiation of heat, as well as allowing the inside wall to expand freely.

The covering of the back connection may be flat or arched. It is generally carried on cast-iron supports of tee section. If arch-bricks are used, those next to the boiler-head can rest on a two-inch angle riveted to back head of boiler. The lower edge of this covering should be below the water-line.

The blow-off should be in the bottom of shell near the rear end. The pipe, if unprotected, is liable to burn off, as there is no circulation through it. It can be protected by covering with pieces of cast iron or hard-burned tile pipe, slipped over, as shown in Fig. 12. By connecting this blow-off pipe with a branch entering the boiler near the water-line a circulation can be maintained which will materially assist in preventing injury, but such a connection adds extra joints and parts that may give trouble. When properly made, this circulating connection can be commended.

The tubes are often too close to the shell. They should be spaced from the sides of shell far enough to allow a generous down-current water-space. The distance from side of shell to outside of nearest tube should not be less than 4 inches, and a vertical line drawn from the point where water-line strikes the side of shell should pass outside the top row of tubes. The tubes should be in horizontal and vertical rows, and not be staggered.

There should be a manhole in shell above the tubes, and when possible one in front head below the tubes.

Figs. 10, 11 and 12 show different designs. The good points of one may be embodied in another to suit requirements.

The Upright or Vertical Boiler is a very useful form where economy of floor-space is a requisite. The small sizes are easily portable, and no setting is required (Figs. 13 and 14).

When the size will permit, they should be made of one sheet, thus having one vertical seam, which should be double-riveted, and two ring-seams at the ends, which may be single-riveted. When more than one sheet must be used, the lap on the ring-seam should be downward on the inside, so as not to obstruct the downward current of water or form a ledge to catch sediment.

TABLE XI
HORIZONTAL RETURN-TUBULAR BOILERS
A List of Commercial Sizes*

Horse-power	30	35	40	45	50	60	70	75
Heating surface, in square feet.	429	506	568	618	684	800	944	1010
Diameter of boiler, in inches.	42	44	48	50	54	54	60	60
Length of boiler, in feet and inches. . .	13'	13'	13' 2"	14' 2"	14' 2"	16' 2"	15' 4"	16' 4"
Thickness of shell. . .	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$
Thickness of head. . .	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$
Length of tubes, feet	12	12	12	13	13	15	14	15
No. of tubes 3" in diameter.	38	46	52	52	58	58	76	76
No. of tubes 3½" in diameter.				42	50	50	60	60
No. of tubes 4" in diameter.				37	40	40	50	50
Diameter of nozzles, in inches.	3	3	4	4	4	4	5	5
Length of furnace, in feet.	4	4	4	4	4½	5	5	5
Diameter of stack required, in inches. .	20	22	24	24	26	26	28	28
Wt. of bare boiler, about.	5,000	5,500	6,400	6,800	7,300	8,550	10,000	10,500

Horse-power	80	90	100	125	150	175	200
Heating surface, sq. ft. . .	1080	1249	1350	1674	1829	2115	2511
Diameter of boiler, inches	60	66	66	72	72	78	84
Length of boiler, in feet and inches	17' 4"	16' 5"	17' 5"	17' 6"	19' 6"	19' 6"	19' 8"
Thickness of shell	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$
Thickness of head	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$
Length of tubes, in feet. . .	16	15	16	16	18	18	18
No. of tubes 3" in diam. . .	76	96	96	122			
No. of tubes 3½" in diam. .	60	72	72	96	96	112	136
No. of tubes 4" in diam. . .	50	60	60	74	74	90	108
Diameter of nozzles, in ins.	5	6	6	6	6	7	8
Length of furnace, in feet. .	5	5½	5½	6	6' 6½"	6' 6½"	7
Diameter of stack required, in inches	28	32	32	36	38	40	42
Weight of bare boiler, about	11,200	13,300	14,000	17,500	18,500	21,000	25,000

* From Catalogue of the Bigelow Co., New Haven, Conn.

The upper head is flanged to meet the shell. At the bottom the shell and fire-box are united through a mud-ring of forged iron or steel, or the fire-box is flanged below the grate to meet the shell.

The fire-box is internal and the crown-sheet forms the lower tube-sheet. The tubes support and stay the crown-sheet and upper head.

The fire-box sides are stayed to the shell by bolts screwed through both sheets and riveted over. Sometimes these stay-bolts are protected by sockets, made of pieces of pipe cut just the length between boiler-shell and fire-box.

The fire-box sheet should be made as thin as the spacing of the bolts will permit, so as to prevent burning and to transmit the heat freely. The mud-ring should be deep enough to overcome the tendency to turn, due to the greater expansion of the tubes and fire-box over that of the shell.

The width of the water-leg should be as wide as convenient in order to reduce the bending of the stay-bolts as much as possible, due to the greater expansion of the fire-box sheet. The width should never be less than $2\frac{1}{2}$ inches in the smallest boilers, and $2\frac{3}{4}$ or 3 inches is preferable.

The tubes are generally two inches in diameter but may be

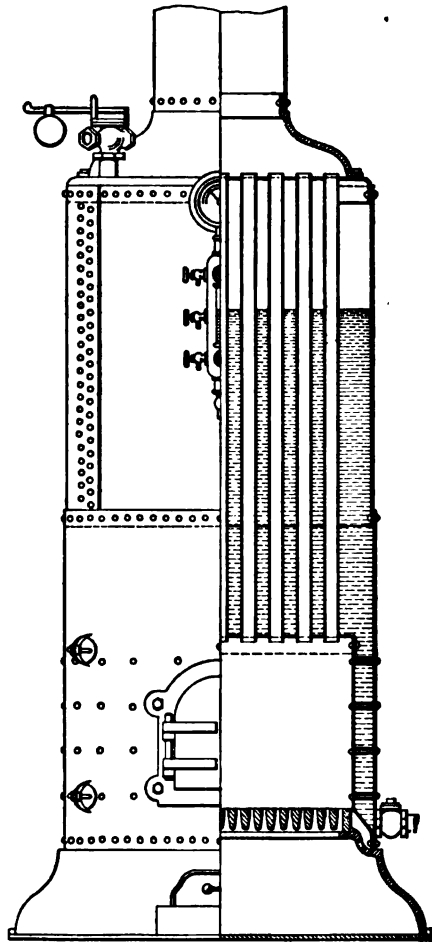


FIG. 13.—Upright or Vertical Boiler.

increased with the size of the boiler. In the ordinary form the upper ends of the tubes are in the steam-space and subjected to extreme heat, which may cause injury. In consequence the tubes are sometimes made totally submerged by having the upper tube-sheet below the water-line, and connecting it with a smoke-flue to the upper head, as in Fig. 14.

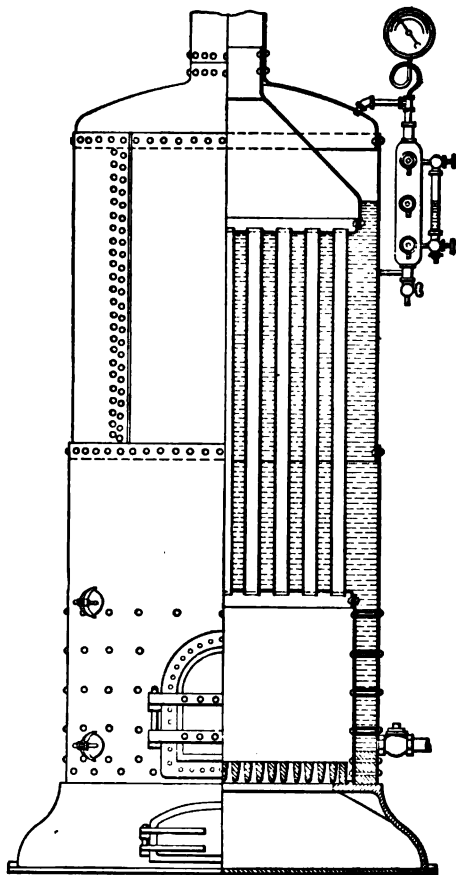


FIG. 14.—Upright or Vertical Boiler with submerged tube-sheet.

There should be hand-holes in the water-leg just above the mud-ring for cleaning out sediment. It is convenient to place a chain at the bottom of the water-leg which can be worked around through the hand-holes to assist in removing scale and dirt. There should be another hand-hole at the level of the crown-sheet, so placed as to reach all parts for cleaning and inspection.

The height of the fire-box should be as great as possible, but not less than 20 inches from top of grate in boilers 24 inches diameter, and 36 inches

in boilers 60 inches diameter.

The boiler usually rests on a cast-iron base, which forms the ash-pit.

The Manning Boiler is one of the best-known types of large size vertical tubular boilers (Figs. 15 and 15a).

These boilers are set on a brick foundation, forming the ash-pit,

TABLE XII
VERTICAL TUBULAR BOILERS
A List of Commercial Sizes

Horse-power.....	3	5	6	7	8	10	12	15
Heating-surface, in sq. ft. . .	39	58	72	84	100	120	144	180
Diameter of boiler, inches. . .	21	24	27	30	30	32	34	36
Height of boiler, feet.	5	5	5	5	6	6	6½	7
Diameter of furnace, inches. . .	16	19	22	25	25	27	28½	30½
Height of furnace, inches. . .	24	24	24	24	24	24	24	24
Thickness of shell.	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Thickness of furnace.	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Thickness of heads.	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Number of 2-inch tubes. . . .	20	31	39	46	46	51	56	64
Length of tubes, inches. . . .	36	36	36	36	48	48	54	60
Diameter of bottom of base, inches.	27	31	35	37	37	39	41	43
Height of base, inches.	11	11	13	13	13	13	13	13
Height of bonnet, inches. . . .	8	9	10	11	11	12	13	13
Height of boiler, bottom of base to top of bonnet, inches.	79	80	83	84	96	97	104	110
Height of boiler, bottom of wheels to top of bonnet, inches.	91	92	95	96	108	109	116	122
Diameter of smoke-stack required, inches.	7	8	9	10	10	11	12	14
Weight of boiler without fixtures.	750	900	1100	1300	1450	1540	1750	2100
Weight of fixtures.	300	400	425	500	500	600	650	700

Horse-power.....	18	20	25	30	35	40	50
Heating surface, in square feet. . .	212½	238	292	358	412	480	578
Diameter of boiler, inches.	40	40	44	48	48	48	54
Height of boiler, feet.	7½	7½	8	8	8	9	9
Diameter of furnace, inches. . . .	34½	34½	38½	42½	42½	42½	48½
Height of furnace, inches.	30	30	30	30	30	30	36
Thickness of shell.	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$
Thickness of furnace.	No. 2	No. 2	No. 2	No. 2	No. 2	No. 2	No. 2
Thickness of heads.	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
Number of 2-inch tubes.	74	84	95	118	138	138	178
Length of tubes, inches.	60	60	66	66	66	78	72
Diameter of bottom of base, ins. .	47	47	50	54	54	54	60
Height of base, inches.	13	13	13	13	13	13	13
Height of bonnet, inches.	13	13	16	16	16	16	18
Height of boiler, bottom of base to top of bonnet, inches.	116	116	125	125	125	137	139
Height of boiler, bottom of wheels to top of bonnet, inches.	128	128	137	137	137	149	151
Diam. of smoke-stack required, inches.	16	16	17	18	18	18	22
Weight of boiler without fixtures. .	2600	2700	3500	4000	4250	4850	5550
Weight of fixtures.	900	900	1000	1050	1050	1050	1200

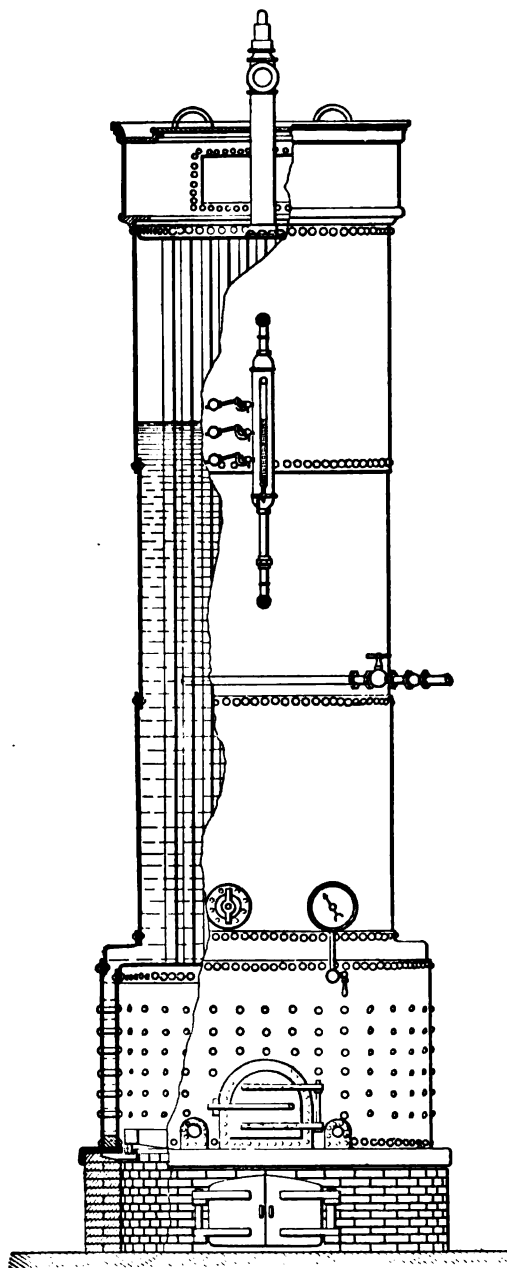


FIG. 15.—Manning Vertical Boiler.

which consists of a circular wall $12\frac{1}{2}$ inches thick, having the inside diameter the same as that of the fire-box. In order to provide increased grate area as well as to allow for expansion, the shell is enlarged by a double-flanged throat-piece just above the top of the combustion-chamber.

The tubes are generally $2\frac{1}{2}$ inches diameter and from 12 to 15 feet long. They are arranged in four sections, so as to leave two cleaning-spaces, like the arms of a cross. Opposite these spaces handholes are located to reach all parts of the top of the crown-sheet.

The feed-water is introduced through a perforated pipe, located near the middle of the boiler.

The Flue and Return-tubular Boiler makes a very convenient form of

internally fired boiler for stationary work (Figs. 16, 16a and 16b). It requires no setting beyond the saddles, which may be either of cast iron or steel built up with angles and plates. The saddle is curved to fit the shell and is about one-third the diameter in length.

It is similar to the Scotch boiler, but has an external back connection for draft between the flue and tubes, in place of an internal one. This back connection is lined with fire-brick. There may be

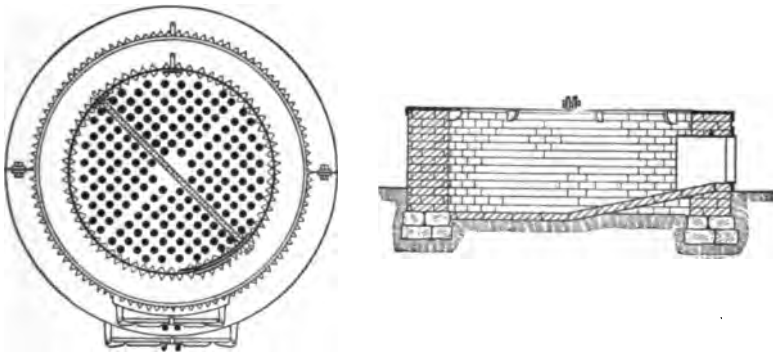


FIG. 15a.—Manning Vertical Boiler. Sections of Fig. 15.

one or two furnace-flues to suit the diameter of shell. The length of grate should not exceed twice the diameter of flue.

These boilers are economical, suitable for mechanical draft (especially the induced draft), and are entirely self-contained.

The Cornish Boiler consists of a cylindrical shell with one large flue (Figs. 17 and 17a). This flue has a diameter of about one-half that of the shell, and is so placed as to leave $4\frac{1}{2}$ or 5 inches between it and the nearest part of the shell. On the continent of Europe this flue is often placed on one side with the object of increasing the circulation, but no material advantage is noticed. This flue is built up of short lengths so as to keep its thickness as thin as is consistent with strength. It is frequently strengthened by Galloway tubes, as shown in Figs. 17 and 45. These tubes are usually about 10 or 11 inches in diameter at the top, and one-half of that at the bottom end, and provide additional effective heating surface.

The boiler is fired internally; the gases pass through the flue to the back end, then under the boiler and back again on each side,

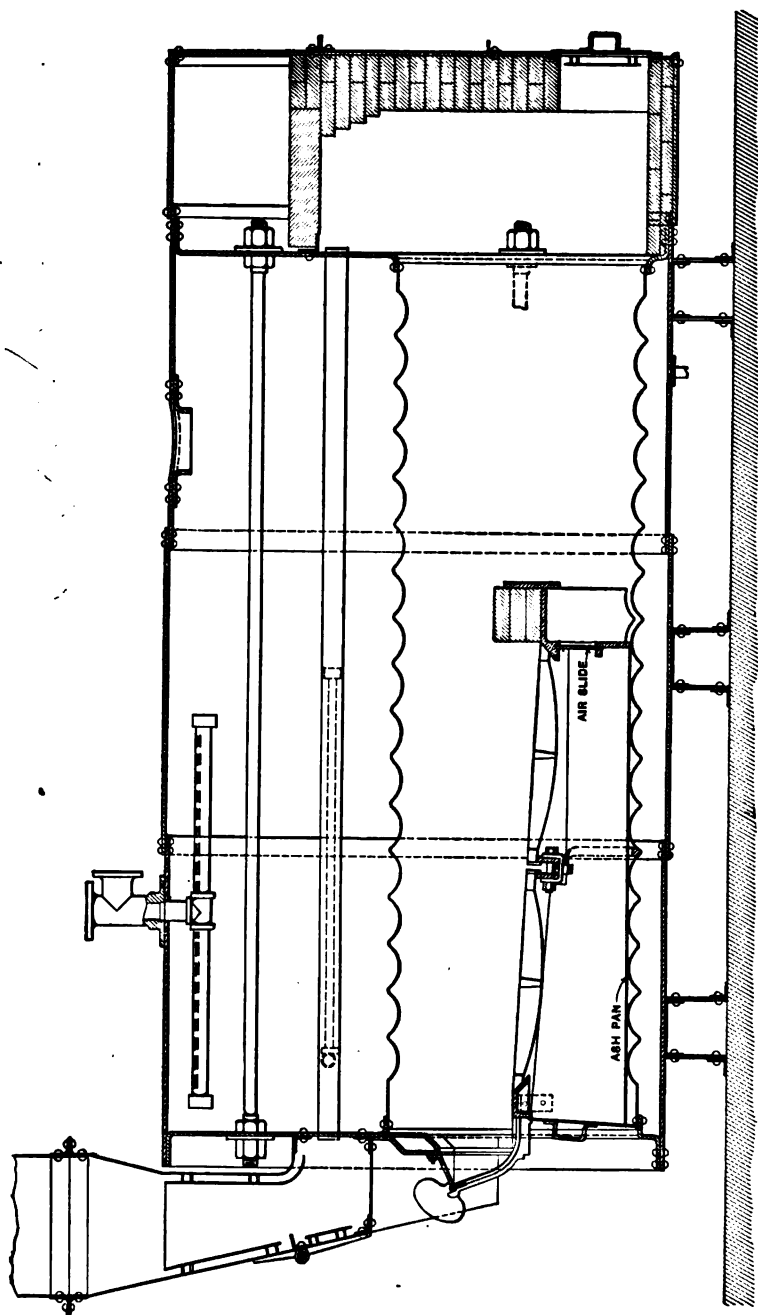


FIG. 16.—Flue and Return-tubular Boiler.

making a "split-return draft." At other times the gases pass to the front on one side and back on the other, making a "wheel draft."

The boiler is set in brickwork. The back head is usually flanged in to meet the shell, while the front head is fastened with an angle placed on the outside of the shell. This arrangement allows a certain amount of spring in the head and thus provides for the expansion of the flue.

As all parts are easily cleaned, this boiler can be used with hard waters, although the large flue and lack of heating surface are decided drawbacks. In consequence Cornish boilers are not used as much as formerly, and are not as favorably received as the Lancashire.

The Lancashire Boiler is similar to the Cornish, but has two flues instead of one (Figs. 18 and 18a). These flues have a diameter of

about one-third that of the shell, and are placed so as to leave about 4 or $4\frac{1}{2}$ inches between them and between the flues and shell. The strengthening rings on the flues can be spaced so as to stagger and thus leave more room for cleaning.

Common sizes for these boilers are 7 feet 6 inches diameter by 30 feet long, with flues each 36 inches diameter; or 8 feet diameter by 33 feet long, with flues each 39 inches diameter. The usual

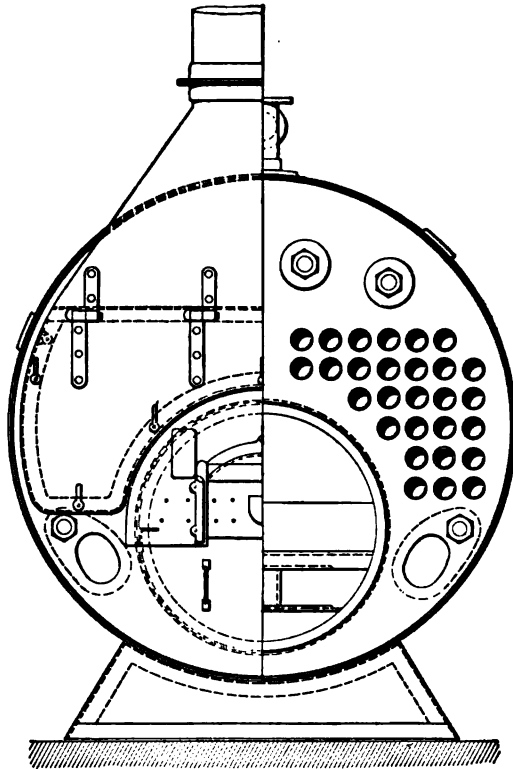


FIG. 16a.—Flue and Return-tubular Boiler.
Front view and section of Fig. 16.

pressure is 100 pounds on the square inch, although pressures as high as 125 pounds are not uncommon. When the flues are fitted with Galloway tubes 160 pounds is sometimes used. These boilers are occasionally fitted with three smaller flues, and are then known as "three-flue Lancashire boilers." When the furnace-flues unite

into one large flue, strengthened with Galloway tubes, they are called Galloway boilers (Figs. 19 and 20).

The setting is of brick. The shell may be supported on cast-iron saddles, as in Fig. 18; or on fire-clay seating blocks, as in Figs. 17 and 20. These blocks have a bearing surface about 5 inches wide, and extend the full length of the boiler. The shell may be carried on one cast-iron saddle, as in

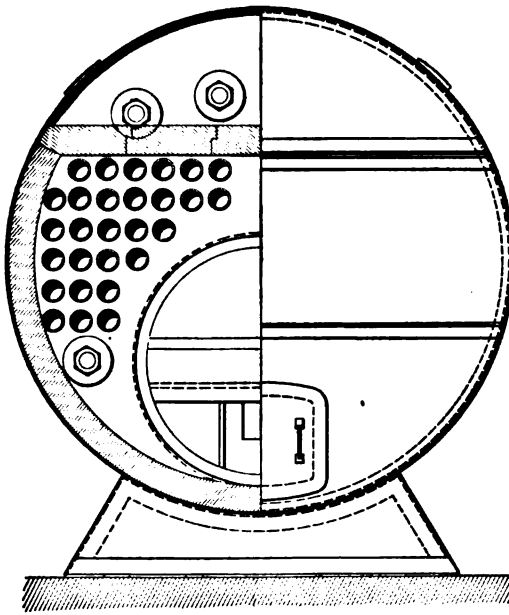
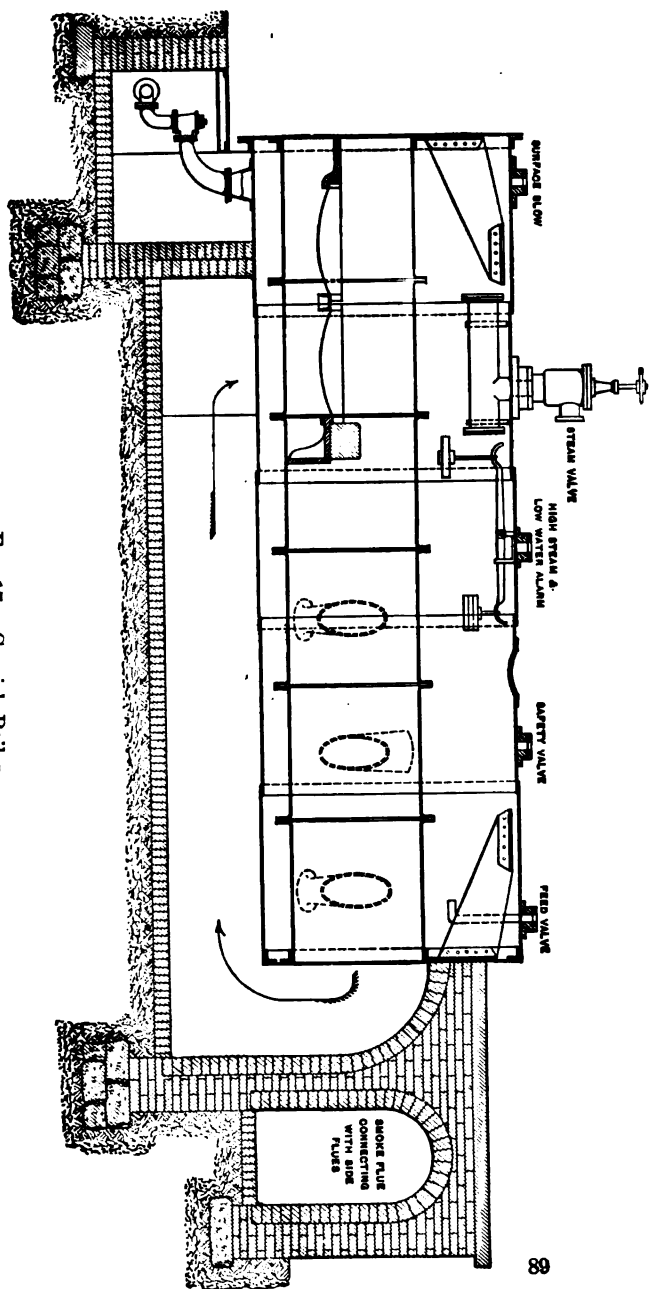


FIG. 16b.—Flue and Return-tubular Boiler.
Back view of Fig. 16.

Fig. 19. The front end is then supported on the brickwork, while the rear end is carried on the saddle made of three parts to allow play for expansion—a foot-block, a saddle to fit the shell, and an intermediate rocker. There is also a brick safety-wall, which deflects the hot gases from the iron support. The draft may pass like that of the Cornish boiler or be made to return over the top.

Any of the settings illustrated in Figs. 17, 18, 19, and 20 apply alike to Galloway, Cornish and Lancashire boilers, and any form of strengthening for the flues may be adopted.

The flues are sometimes reduced in diameter at the rear end to facilitate removal and give sufficient clearance from shell to allow



the back head to spring, which head must be fastened to the shell by an internal flange or angle. The former method is much the

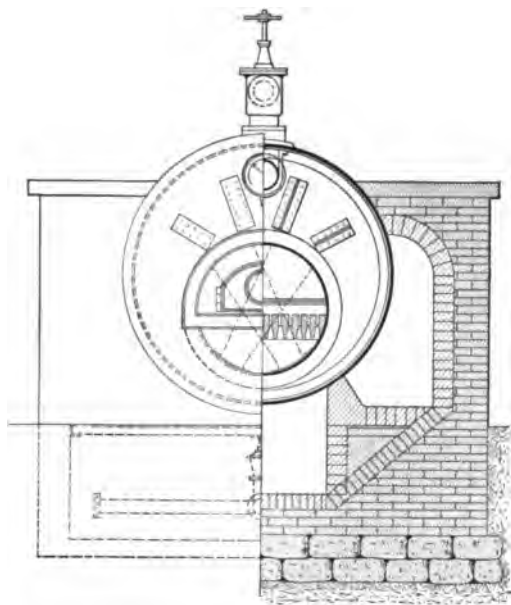


FIG. 17a.—Cornish Boiler.
Front view and section of Fig. 17.

better as being stronger and less stiff. An outside angle, like that on front head, cannot be used, as it would interfere with the draft-current and rapidly burn away.

The Lancashire boiler is an economical type and is well liked in Europe. Best results are obtained when used with an economizer or feed-water heater placed in the flue leading to the chimney, especially if high rates of combustion are adopted.

It requires a large floor-space, and has been but little used in America, chiefly on that account.

The bottom blow-off connection is customarily made at the front end. Great care should be taken not to conceal it in the brick setting, as accidents have occurred from undiscovered corrosion of this part.

The low-water and the dead-weight safety-valves shown in the illustrations are not necessarily a part of the design, but are commonly adopted in foreign practice.

The Scotch or Drum Boiler consists of a cylindrical shell, internally fired in large furnace-flues, the gases returning through a back connection, or combustion-chamber, and a bank of tubes. It may be single-ended as in Figs. 21 and 22, or double-ended as in Fig. 23.

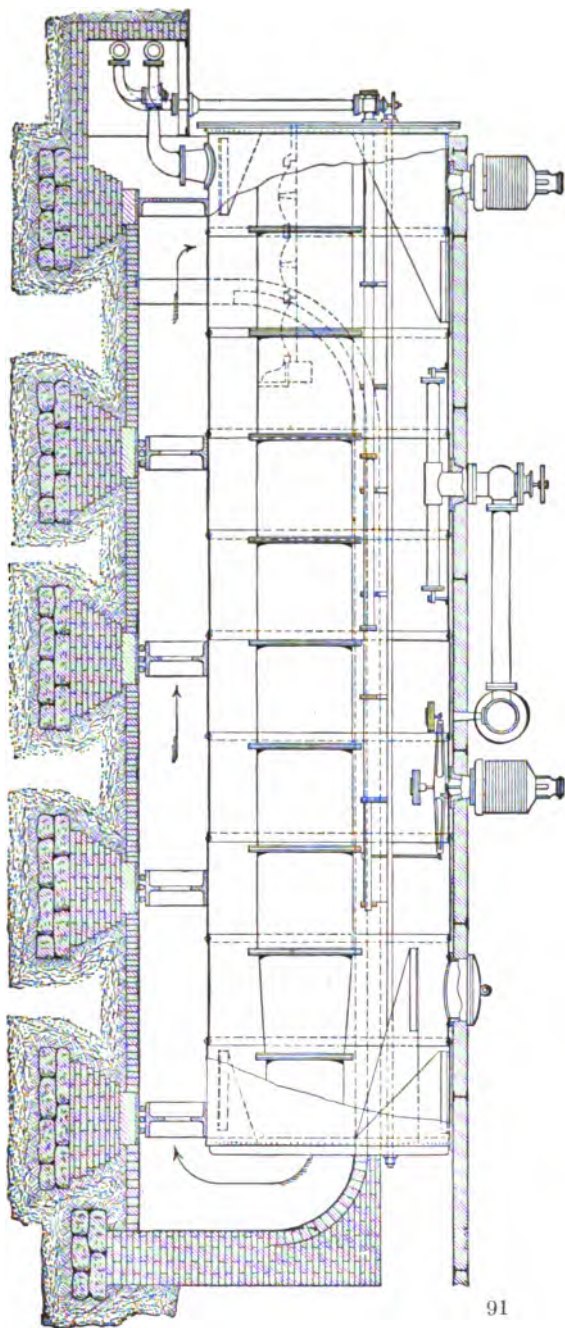


FIG. 18.—Lancashire Boiler.

It is principally used in marine work, where it is much liked on account of its reliability, but can be adopted for stationary practice.

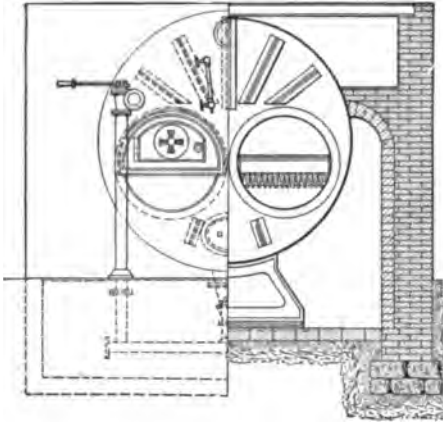


FIG. 18a.—Lancashire Boiler.
Front view and section of Fig. 18.

It is necessarily heavy, due to the thickness of metal and large amount of water contained; and strong efforts have been made, coincident with the increase of steam-pressures, to adopt other forms. It is entirely self-contained; is supported on saddles; is very economical; and, when properly designed, is not difficult to clean or repair.

There may be one, two, three or four furnaces, or twice that number if double-ended. It is a frequent fault to use furnaces of too small a diameter. Two large furnaces are better than three small ones, or three large ones than four small. Furnaces less than 33 inches in diameter cramp the area over bridge wall and the height of ash-pit, thus restricting the draft and chilling the gases with the low crown. The back connection or combustion-chamber may be common to all furnaces or be divided, one to each. With four furnaces it is usual to fit two combustion-chambers, each common to two furnaces. Separate combustion-chambers are to be preferred with mechanical draft. When double-ended there should be separate combustion-chambers for each end.

The tubes should be arranged in nests, leaving a clear vertical space between the banks of tubes and between the tubes and shell. These spaces should be about twice the pitch of tubes. The distance from the side of shell to the outside of nearest tube should not be less than 8 inches, and a vertical line from the point where the water-line strikes the inside of the shell should pass outside of the two top rows of tubes, otherwise the boiler is liable to prime. The uppermost row of tubes should be at least 0.3 of the boiler diameter below the top of the shell, in order to obtain proper water-

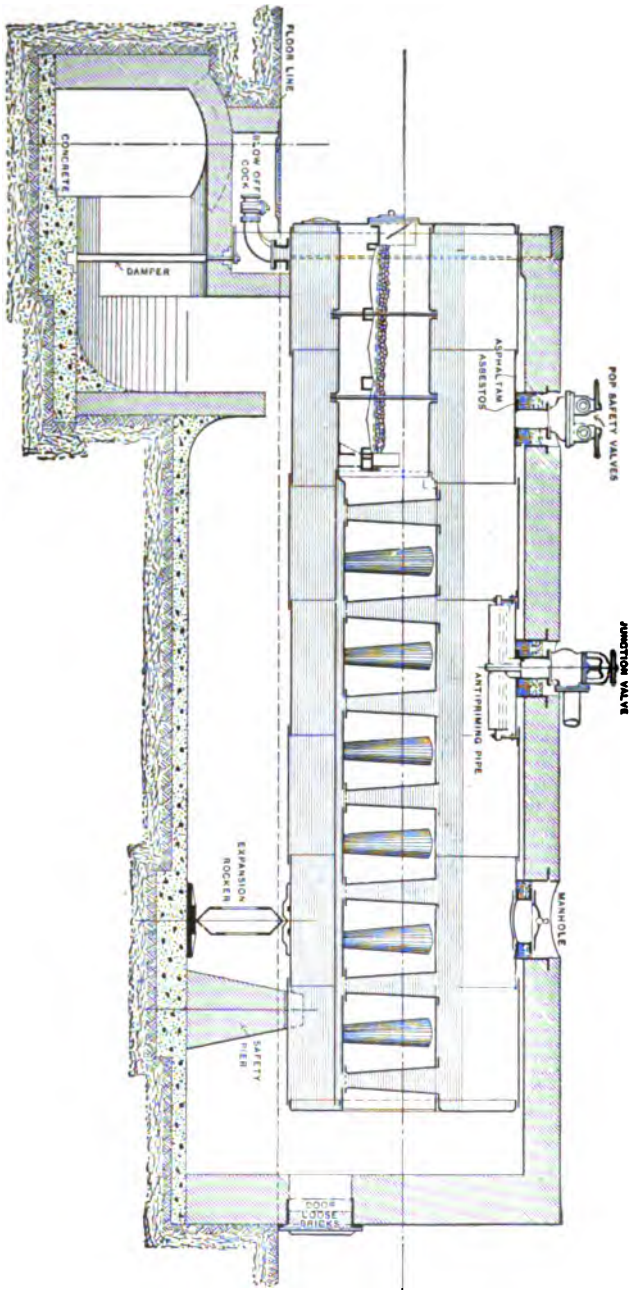
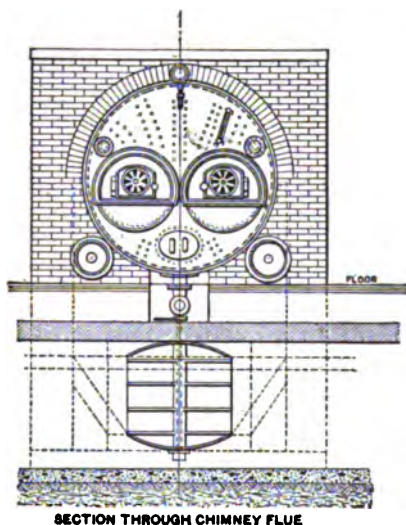


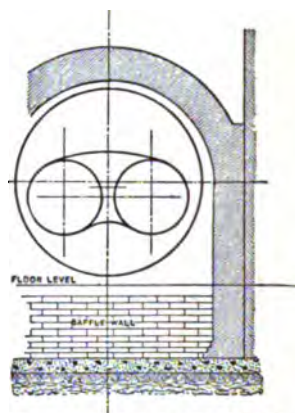
FIG. 19.—Galloway Boiler with expansion rocker support.

separating surface and to prevent priming. The tubes should be always arranged in vertical and horizontal rows, and not be



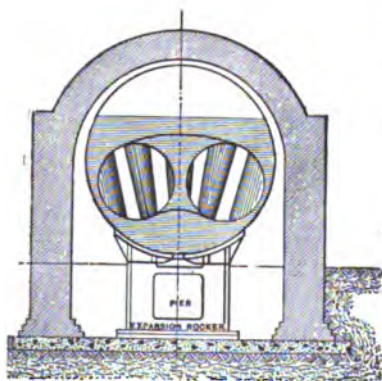
SECTION THROUGH CHIMNEY FLUE

FIG. 19a.—Section of Galloway Boiler, Fig. 19.



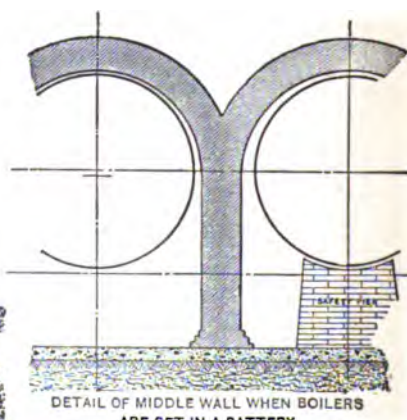
DETAIL OF WALLS WHEN BOILER IS SET AGAINST WALL OF BUILDING

FIG. 19b.—Section of Galloway Boiler, Fig. 19.



CROSS SECTION—ONE BOILER SET ALONE.

FIG. 19c.—Section of Galloway Boiler, Fig. 19.



DETAIL OF MIDDLE WALL WHEN BOILERS ARE SET IN A BATTERY

FIG. 19d.—Section of Galloway Boiler, Fig. 19.

staggered. The horizontal spacing is generally made wider than the vertical, to facilitate the rising currents carrying the steam-

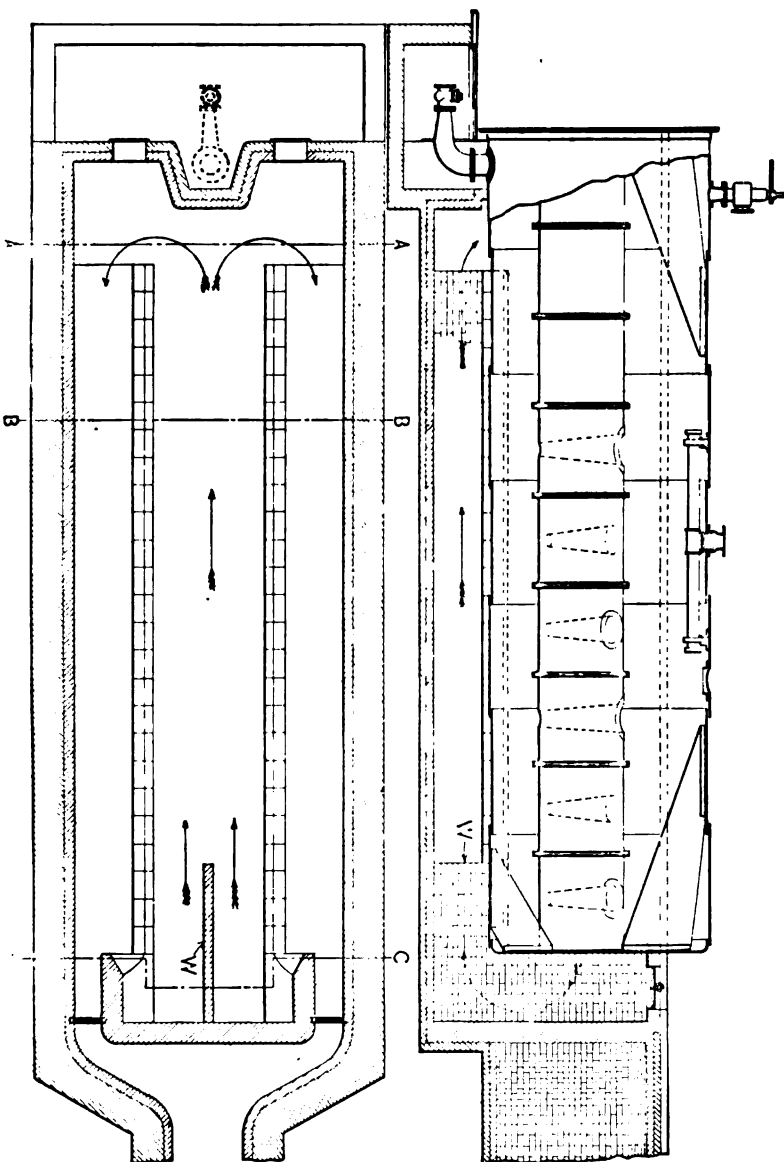


FIG. 20.—Lancashire Boiler with Galloway Tubes, seated in the type of brick setting most used. For Galloway and Cornish boilers the division wall "W" is omitted.

bubbles. When boilers are set fore and aft on shipboard, some designers arrange the tubes so that the rows slope from the centre



FIG. 20a.—Lancashire Boiler. Front Elevation.

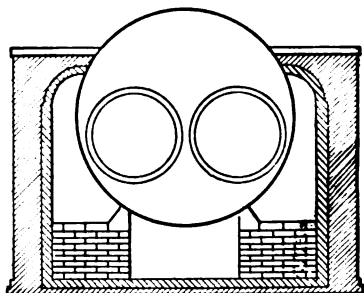


FIG. 20b.—Lancashire Boiler. Section at AA.

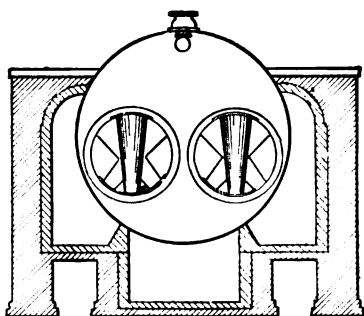


FIG. 20c.—Lancashire Boiler. Section at BB.

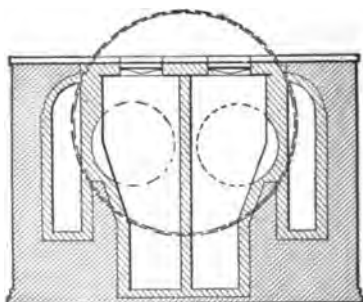


FIG. 20d.—Lancashire Boiler. Section at CC.

toward each side, that they may not be lifted out of the water when the vessel rolls in a seaway.

In order to lighten the boiler by relieving it of the necessarily heavy through-stays in the steam-space, the heads above the tube-line are sometimes curved inward over a long radius (Fig. 22).

It is always best to flange the back end of the furnace to meet the tube-sheet of the combustion-chamber, so as to keep the rivet-heads out of the fire (Figs. 21 and 22). With this arrangement it is difficult to remove a furnace unless it be especially flanged to pass out of the hole in the front head. If the furnace end is straight with the flange on the tube-sheet, the rivets should be half-countersunk.

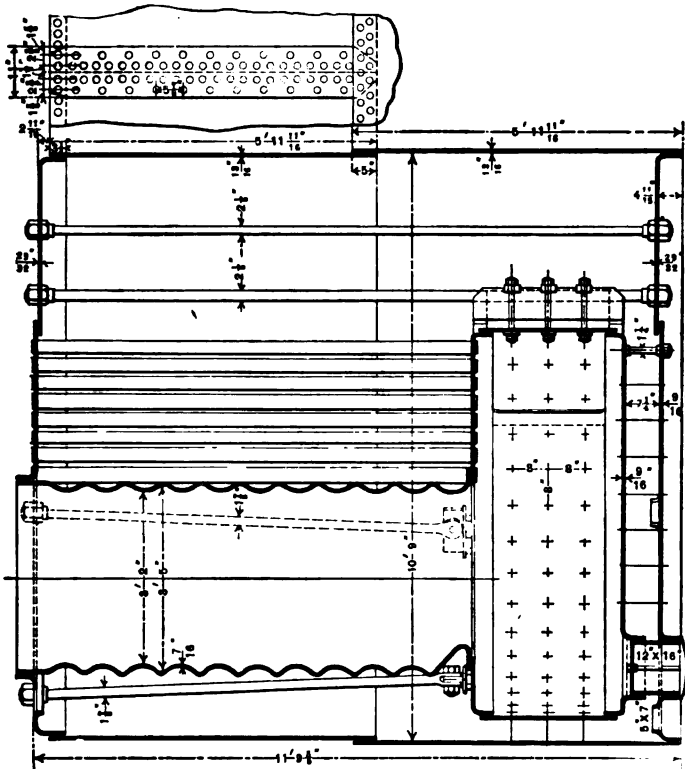


FIG. 21.—Scotch Boiler, single-ended, with common combustion-chamber.

The sides and bottom of the combustion-chamber are made parallel to the shell, and should be spaced away from the shell by not less than 3 inches in the clear, although 4 or 5 inches would be preferable. The back sheet of the combustion-chamber may be parallel to the back head, in which case it should be spaced away by not less than 4 inches in small boilers, nor less than 5 inches in large ones, while 6 inches is to be preferred in all cases. A more expen-

sive but better method is to slope the back sheet away from the back head, leaving a space of about 4 or 5 inches at the bottom and of 8 or 10 inches at the top. The top of the combustion-chamber may be flat, in which case it must be supported. This is done by girder-stays, by crowfoot stays to the shell, or by girders strengthened by stays to the shell. The former method is to be preferred,

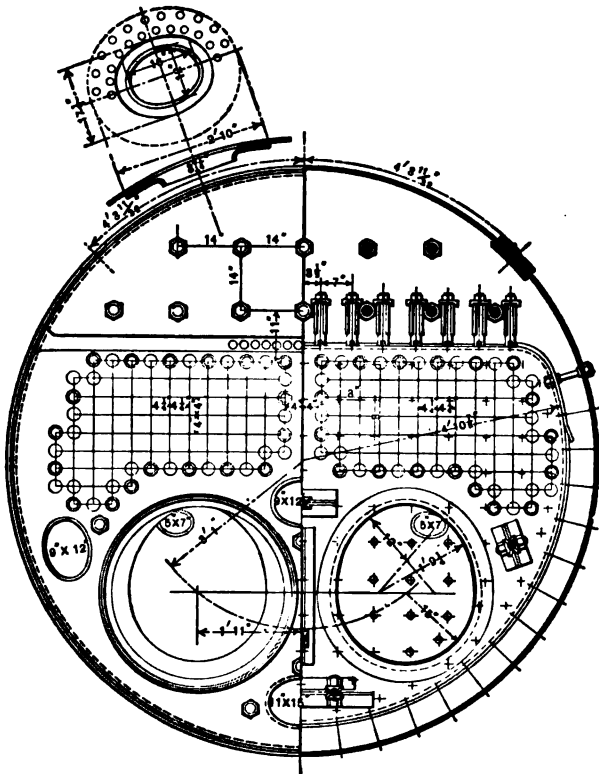


FIG. 21a.—Scotch Boiler. End view and section of Fig. 21.

although crowfoot stays to the shell are cheaper and used in many cases with low pressures. The objection to the combination girder and stay to the shell is the uncertainty that each will carry its calculated stress.

Care should be taken to arrange the longitudinal seams that they are not placed below the furnaces. The water at the bottom

of these boilers is apt to remain cold much longer than that at the top, and seams located in the bottom of the shell are almost sure to occasion annoyance from leaks. In this particular hydrokineters or other devices for creating an artificial circulation are very useful. These attachments consist of an internal nozzle through which steam can be blown from a donkey boiler or other source of supply,

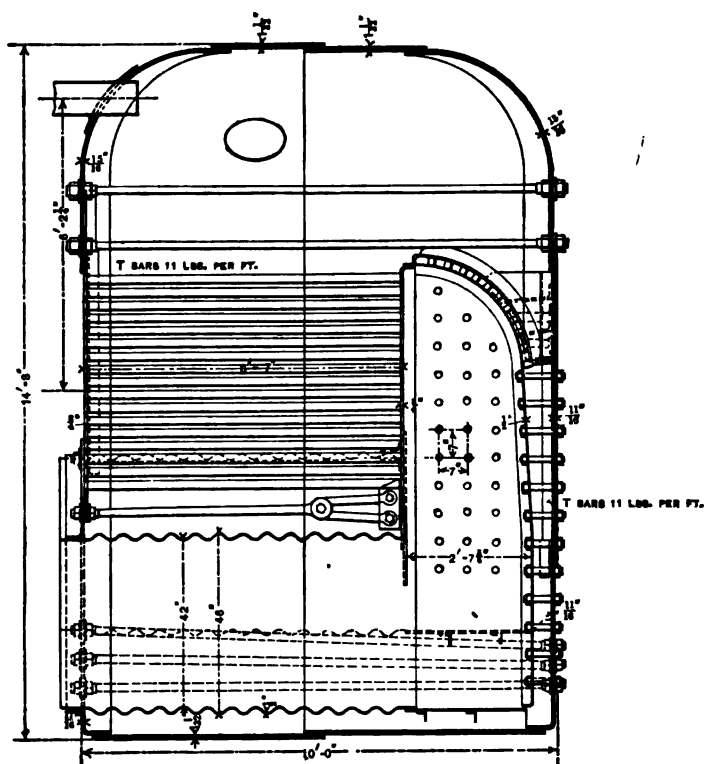


FIG. 22.—Scotch Boiler, single-ended, with separate combustion-chambers.

and thus form circulating currents during the process of generating steam. The cold feed-water should also enter at or near the water-line, so as to assist the natural circulation and help to maintain a more uniform temperature.

The smoke-connection on the front of the boiler may be fastened to an angle riveted or tap-bolted to the boiler-head. The sheets forming the smoke-connection should be arranged so as to cover

the nuts on the ends of the large stays in the steam-space and protect them from the corrosive effect of the gases.

The Admiralty or Gunboat Boiler is a modified form of Scotch boiler especially designed to reduce head-room and still maintain heating surface. This is accomplished by decreasing diameter and increasing length, which necessitates the tubes being placed in line

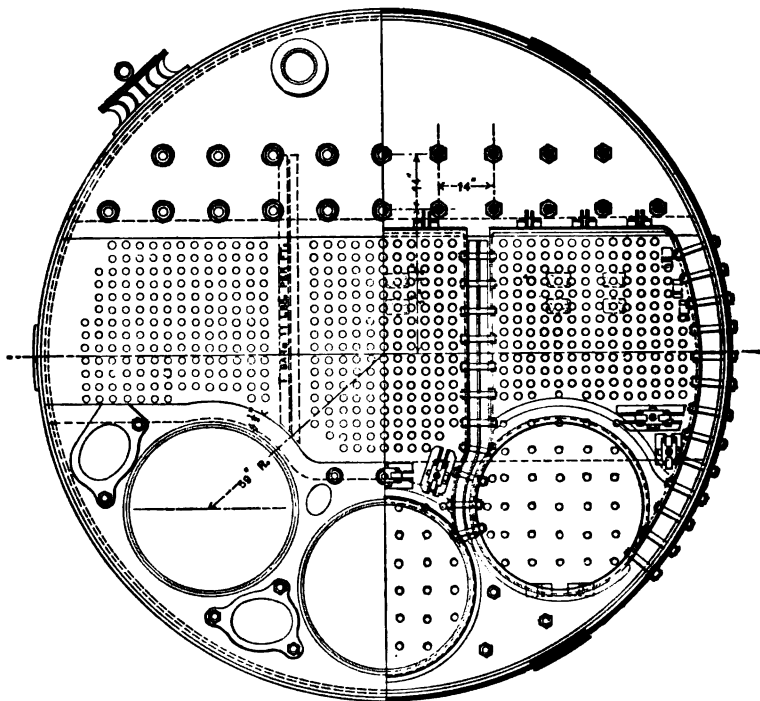


FIG. 22a.—Scotch Boiler. End view and section of Fig. 22.

with the furnace-flues. It has many good features, but requires a long floor-space (Fig. 24).

As the lower tubes are apt to collect all the soot and ashes that are carried over the bridge wall, these tubes are sometimes made one size larger than those above.

The same general comments applicable to the Scotch boiler are true for this type.

The Marine Boiler is the name of a type used on many river and sound steamers in America. It is a very good steaming boiler, but

with pressures exceeding 60 pounds on the inch the flat surfaces are objectionable, and the multiplicity of stays makes it difficult to inspect and clean.

It is internally fired, having a fire-box of the locomotive type, the products passing through flues to a back connection and returning through tubes.

As these boilers are principally used with single-cylinder, long-stroke engines, the steam-space has to be increased by means of a

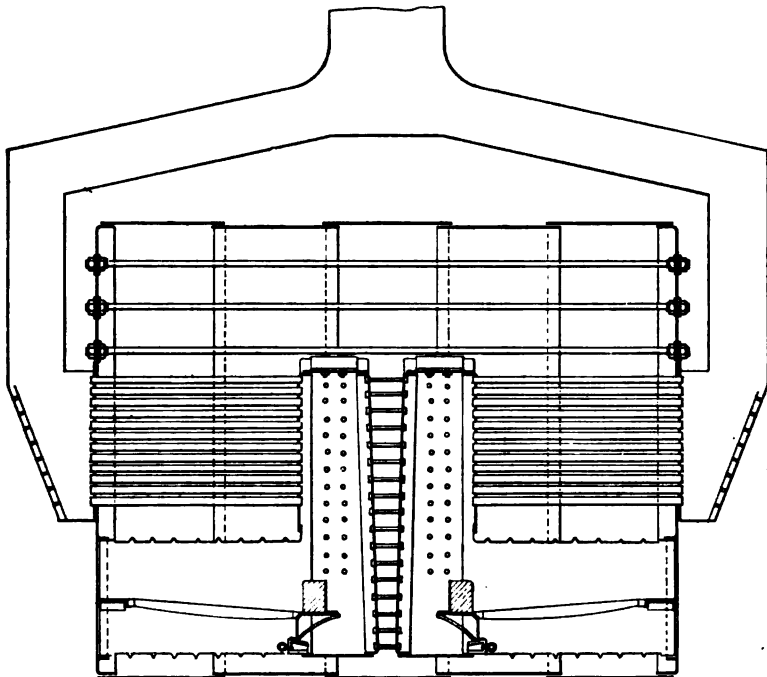


FIG. 23.—Double-ended Scotch Boiler.

steam-chimney (Fig. 25), or by a steam-drum or superheater (Figs. 26 and 27). For high pressures the chimney is objectionable, as it necessitates the cutting out of a large portion of the shell, and in consequence the steam-drum is preferred. Each boiler may have its own steam-drum, or there may be one superheater or drum common to all the boilers. The steam-pipe connecting the boiler to the drum is best arranged to enter the side of drum about one-third

or one-quarter of its height from the bottom. The steam-pipe to the engine should connect to the opposite side and near the top. There should be a small copper drain-pipe to carry the priming and condensed steam from the bottom of the drum back to the boiler. This drain varies usually from 2 to 4 inches in diameter. It should enter the shell of the boiler at or just below the water-line, and does not require a check-valve. The steam-connection to the drum

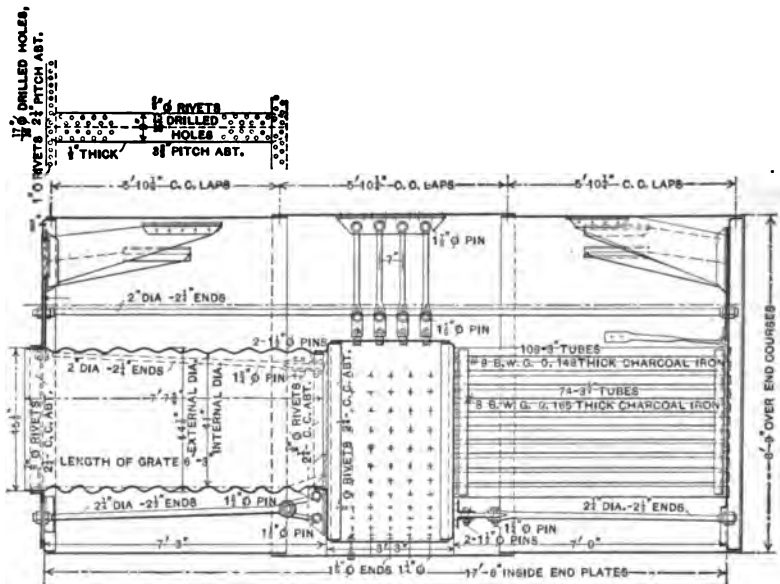


FIG. 24.—Admiralty or Gunboat Boiler.

may enter the bottom, as in Fig. 27, and be extended up on the inside. This particular design was adopted for the sake of avoiding certain obstructions. The side connection, as in Fig. 26, is to be preferred.

The piece forming the bottom of the water-legs of the furnaces should be made out of one sheet, flanged up solid around the corners, or have the corners welded. It should be heavier than the side sheets. The side sheets at the furnace should join the curved top of the shell over the furnace in a longitudinal joint placed not less than 12 inches below the axis of the cylindrical part. The throat-piece, connecting the end of the cylindrical part to the flat

furnace part, is the weakest piece, other things being equal, and requires the most care in fitting and flanging.

The width of furnace end is usually equal to the diameter of cylindrical end (Fig. 25), but may be wider (Fig. 27), to increase

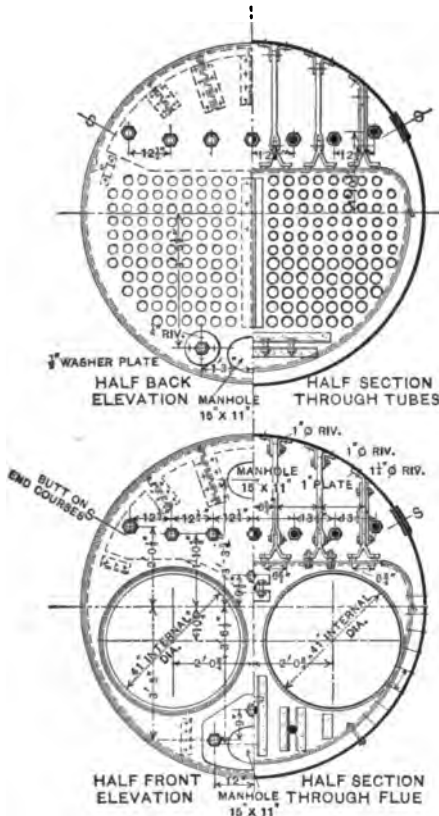


FIG. 24a.—Admiralty or Gunboat Boiler. Sections of Fig. 24.

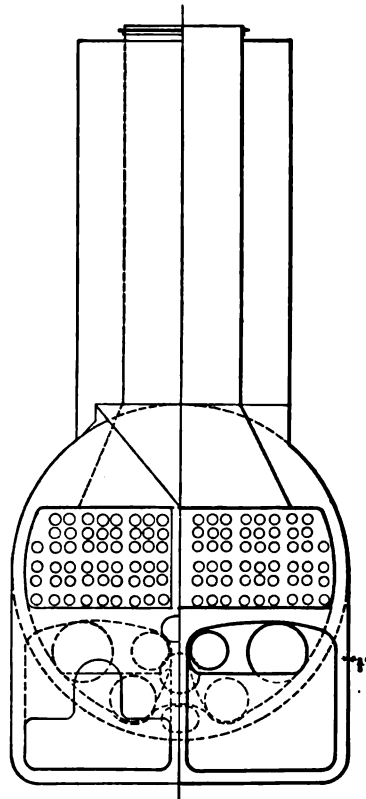


FIG. 25a.—Front end and section of Fig. 25.

the grate surface. The bridge wall may be built of moulded fire-clay or bricks, or be constructed as part of the boiler and then called a "wet bridge."

The boiler is supported on iron saddles cast to fit both the round of the cylindrical part and the base of the water-legs.

The Locomotive Boiler is one of very convenient form and is entirely self-contained. It has been used in a variety of places and

has proved very satisfactory. Its chief objection is the flat sides of the fire-box, necessitating stays, but this objection is true for many other types (Fig. 28).

In locomotive-engine work the tubes are generally made 2 inches in diameter, in order to secure the required heating surface,

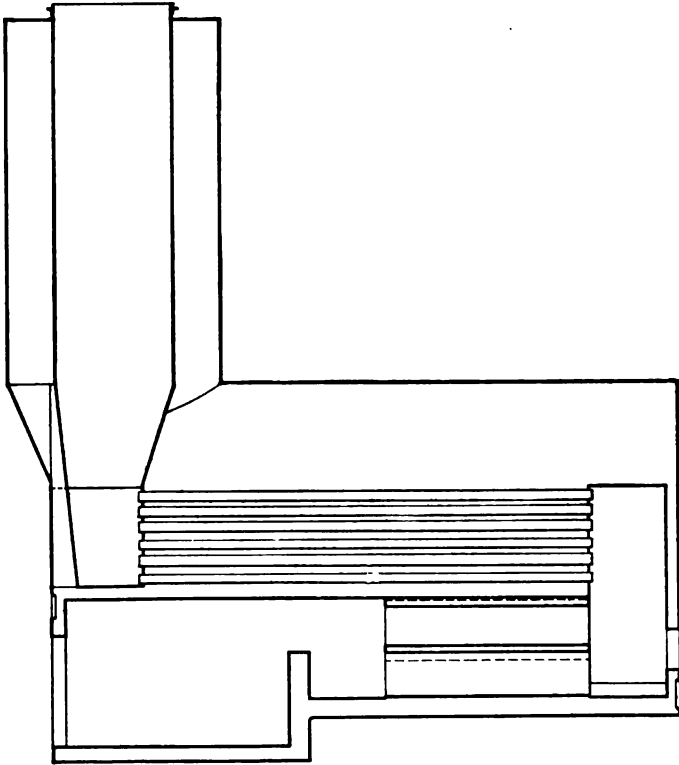


FIG. 25.—Marine Boiler with steam-drum.

and are spaced staggered. The small tubes are permissible on account of the powerful draft produced by the exhaust from the cylinders. The staggering of the tubes does not appear to interfere with the free steaming of the boiler, probably due to the rocking of the machine as it runs on the track, which helps to free the steam-bubbles from the heating surfaces.

The minimum thickness of the tube-sheet is $\frac{1}{4}$ -inch. American practice uses steel for the fire-box side and tube-sheets, but some

foreign engineers still prefer copper. There appears to be less advantage in copper than would naturally be credited to its high conducting power, especially for the tube-sheet. Copper fire-boxes have to be made thicker than steel, and the tube-sheet is additionally thickened in the tube-space so as to give the tubes a firm hold. The water-legs are about 3 inches in the clear.

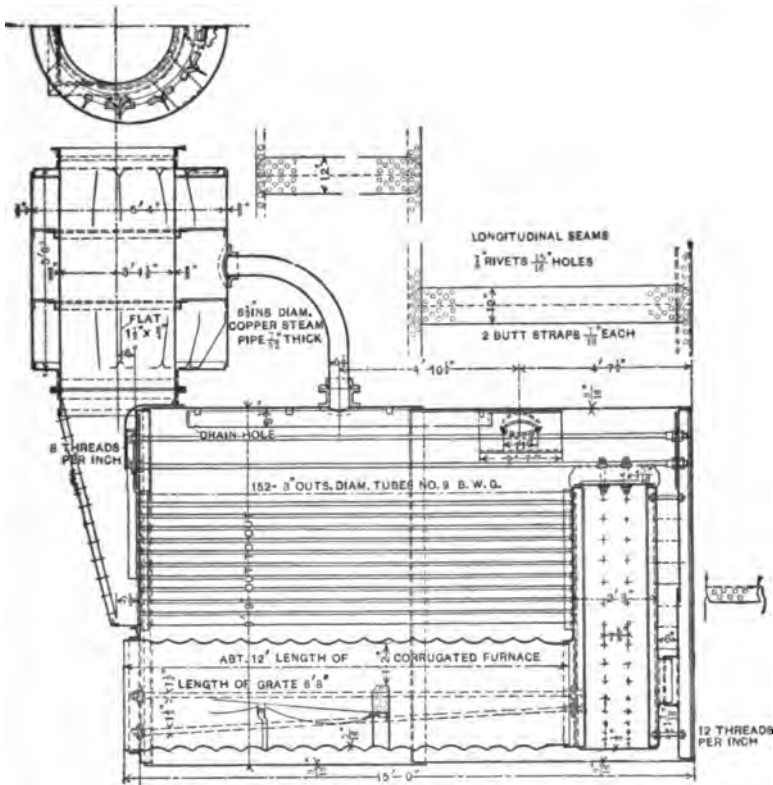


FIG. 26.—Scotch Boiler with steam-drum.

This type is a very convenient one for portable boilers, is self-contained and easily mountable on wheels or skids. These portable boilers are often made with a water-bottom under the ash-pit. The tubes are usually $2\frac{1}{2}$ or 3 inches in diameter.

The Compound Boiler is an attempt to combine the advantages of the fire- and water-tubular types. It has many promising

features, but creates complication and is neither one thing nor the other. At the present time no form has met with what could be termed a commercial success, although some have been reported as giving satisfactory results.

The **Water-tubular Class** of boiler was created by the continued and well-founded demand for high pressures of steam, which neces-

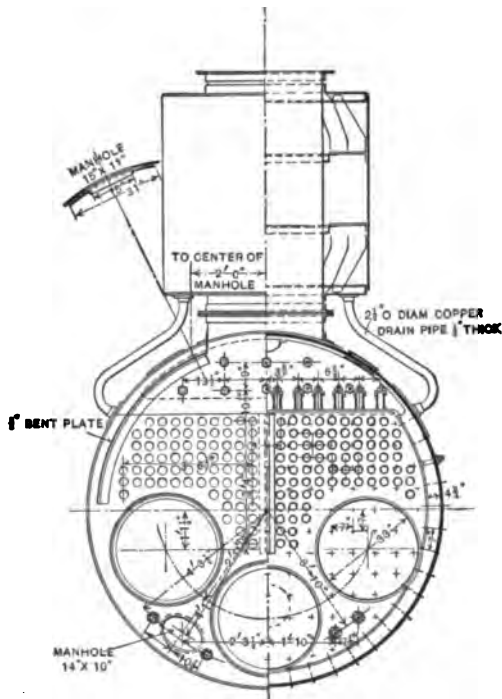


FIG. 26a.—Scotch Boiler.
End view and section of Fig. 26.

sitated parts of great thickness in the older types. In this class of boiler the water is contained in elements of comparatively small size, which reduces the thickness of metal, the quantity of water contained and consequently the total weight of the boiler; and increases the rapidity with which steam can be generated without injury from unequal expansion.

The early attempts were failures, and many of the present designs cannot be commended.

Nearly all the forms, and especially the better ones, are manufactured solely by certain companies under letters patent, and the engineer is not called upon to furnish designs, but simply to determine the selection most suitable for his work.

Literature is full of discussion on water-tubular boilers, and as the subject has not been reduced to fixed conditions, reference is made to articles on the subject already in print. Many of such articles will be found in the Transactions of the naval engineering societies, but those written by manufacturers and interested parties

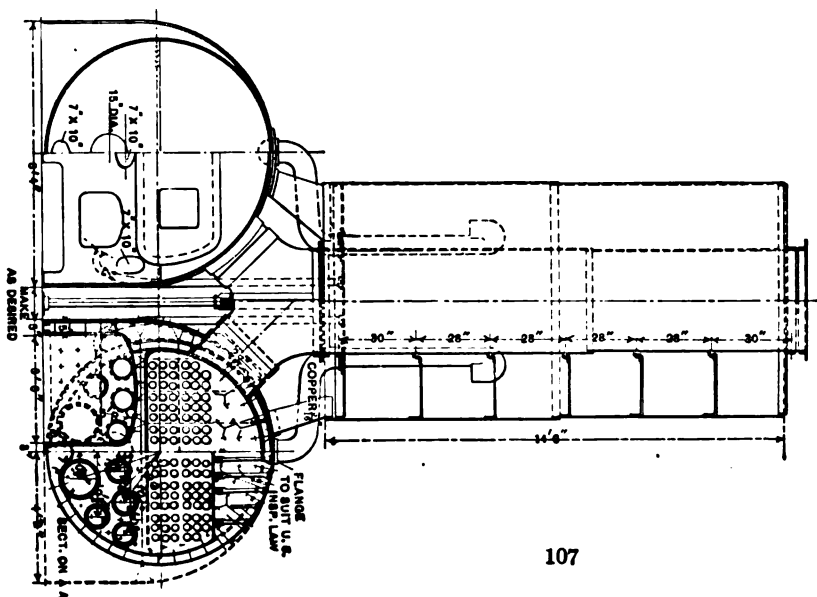
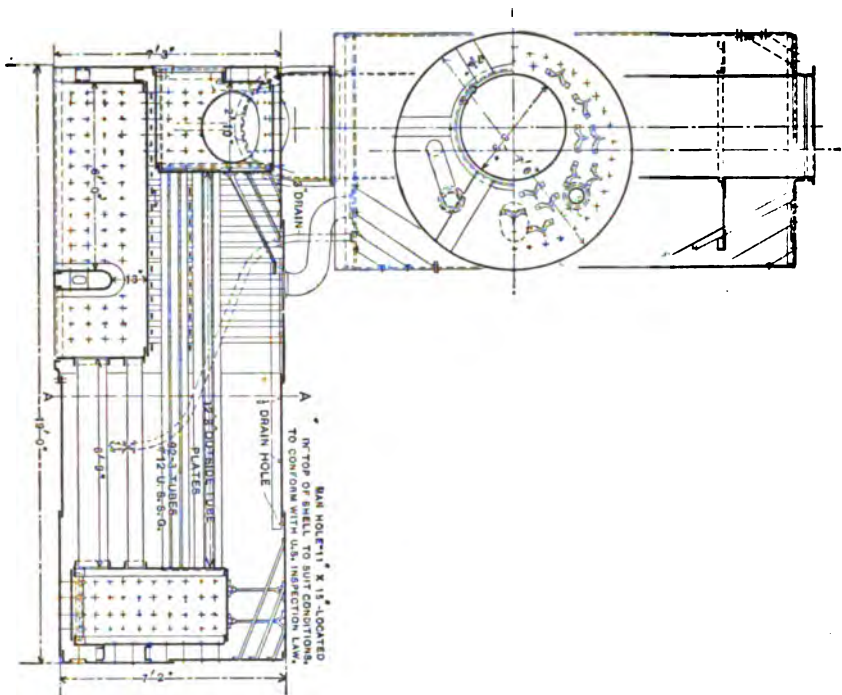


FIG. 27.—Marine Boiler with steam-drum.

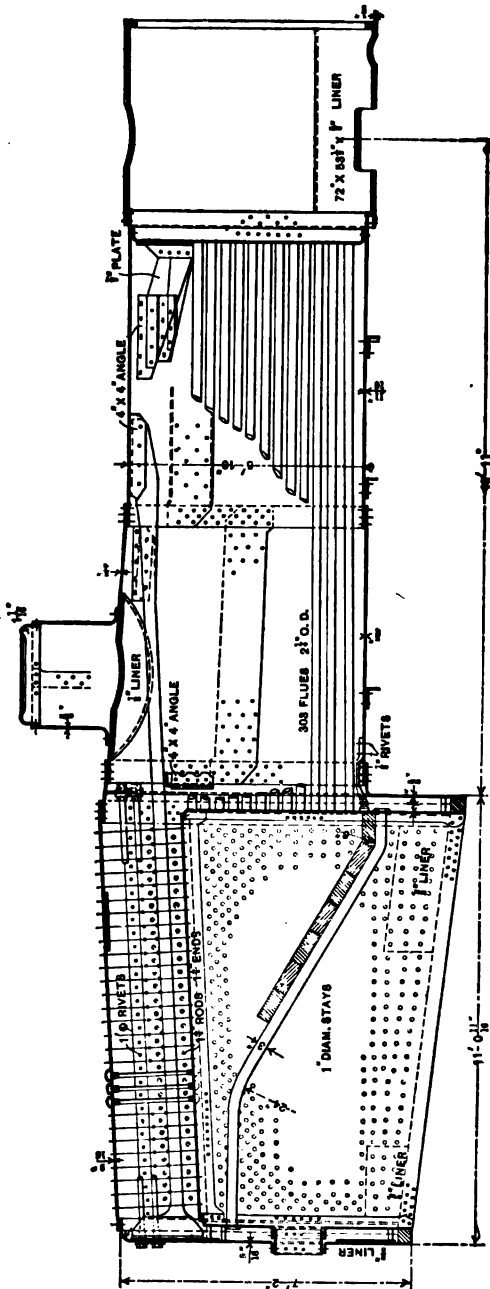


Fig. 28.—Locomotive Boiler.

should be read with some caution. The weights are nearly always underestimated, and the actual weight of boiler, brickwork and setting or casing make the completed boiler much heavier than is usually stated.

The class has been hampered by many poor designs, which have failed and caused distrust, but general condemnation should not be based alone on these experiences.

All things considered, there is no reason why the water-tubular boiler should not be the boiler of the future, and such boilers are in extensive use for both stationary and marine work. They are more complicated, as a general thing, than some of the forms of fire-tubular boilers, and under

best conditions for each have not shown any particular increase in economy. In short, the two types are about equal in efficiency when placed under conditions suitable for each.

The water-tubular class is claimed to be the safer from explosion, due to the lesser amount of water contained. It is no doubt true that explosions with water-tubular boilers are less destructive.

The general principles of boiler construction are as true for water-tubular boilers as for fire-tubular. It does not appear necessary to

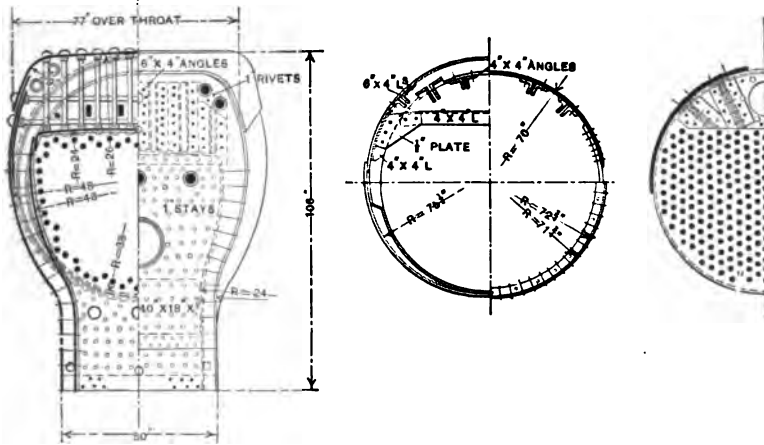


FIG. 28a.—Locomotive Boiler. Sections of Fig. 28.

make the tubes curved or bent in order to allow for expansion, as many straight-tube boilers have proved satisfactory, and straight tubes are certainly simpler and easier to inspect and clean.

Water-tube boilers as a class are capable of being forced to high rates of combustion without serious injury. When so used, however, the tubes burn away faster than the drums and finally give out completely, so that a new set of tubes has to be furnished. The sheet-metal casings are apt to get very hot, even though the casing be lined with some non-conducting material. In consequence considerably more heat is lost by radiation than with the brick-set or internally fired fire-tubular boilers.

The heat-absorbing value of the tube surfaces varies, and no doubt many parts of the same tube are of little value compared to other parts. For good economy in absorbing the heat and reducing

the temperature of escaping gases, the ratio of heating to grate surface should always be large.

As steam can be raised in a much shorter time (about one-eighth to one-tenth) by using a water-tubular boiler, there results a considerable saving in fuel when the boilers have to be frequently started.

A water-tubular boiler, made up with cylindrical surfaces of small diameter, with heads bumped or of such form as not to require staying, is certainly a step toward an ideal form. The multiplicity of parts and joints is an objection, partially offset by the absence of stays. Some of the water-tube forms have departed from the ideal, and nothing has been gained except in the eyes of the maker. The failure of water-tube boilers through unskilled use has raised the question of length of service, but there is lack of positive knowledge to prove that they will not last as long as other forms. It has been shown, however, that water-tube boilers require more constant watching and care than fire-tubular ones, owing to the smaller quantity of water contained and to their sensitiveness to respond to sudden changes of temperature.

A. E. Seaton stated the requirements for a water-tubular boiler before the Institute of Civil Engineers (London, May, 1897) in the following language:

"The ideal boiler referred to—or perhaps preferably the boiler of the future, because it is not likely that any boiler will ever quite fulfil every requirement of an ideal boiler—must have a rapid, uniform and definite circulation, the upcast tubes should be very considerably inclined from the horizontal, and the nearer they are to the vertical position the better; they may be large or small, according to fancy or circumstances; they should be capable of easy examination, and therefore must be straight or nearly so, and their arrangement should be such that any one of them may be easily drawn and replaced; the downcast pipes, or those from the steam-drum to the water-pockets, should be as direct as possible and of considerable size, and at or near their bottoms there should be a receptacle with no circulation—in other words a dead end, so that solid matter can be separated by gravity from the liquid; the fireplace and its surroundings should be of such size and nature as to allow of the proper combustion of the fuel and its effluent gases, while the general structure of the boiler should be such as to

enable it to bear sudden expansion and contraction with impunity, and the whole of the surface exposed to flame and hot gases should be accessible for cleaning. If these conditions are fulfilled, there is no absolute necessity for using pure fresh water, inasmuch as the rapidity of flow will prevent deposition on the upcast pipes by the mechanical scour of the water; the dead ends permit of the deposit at a safe place, and if there is any deposit on the downcast pipes, they—being of considerable size and easy of access—can be cleaned when opportunity serves, and if necessary would go for a considerable period without cleaning.”

George W. Melville, U.S.N., has placed the advantages and disadvantages as follows (Trans. Soc. Naval Arch. and Marine Engs., Nov. 1899):

ADVANTAGES.

- Less weight of water.
- Quicker steamers.
- Quicker response to change in amount of steam required.
- Greater freedom of expansion.
- Higher cruising speed.
- More perfect circulation.
- Adaptability to high pressures.
- Smaller steam-pipes and fittings.
- Greater ease of repair.
- Greater ease of installation.
- Greater elasticity of design.
- Less danger from explosion.

DISADVANTAGES.

- Greater danger from failure of tubes.
- Better feed arrangements necessary.
- Greater skill required in management.
- Units too small.
- Greater grate surface and heating surface required.
- Less reserve in form of water in boiler.
- Large number of parts.
- Tubes difficult of access.
- Large number of joints.
- More danger of priming.

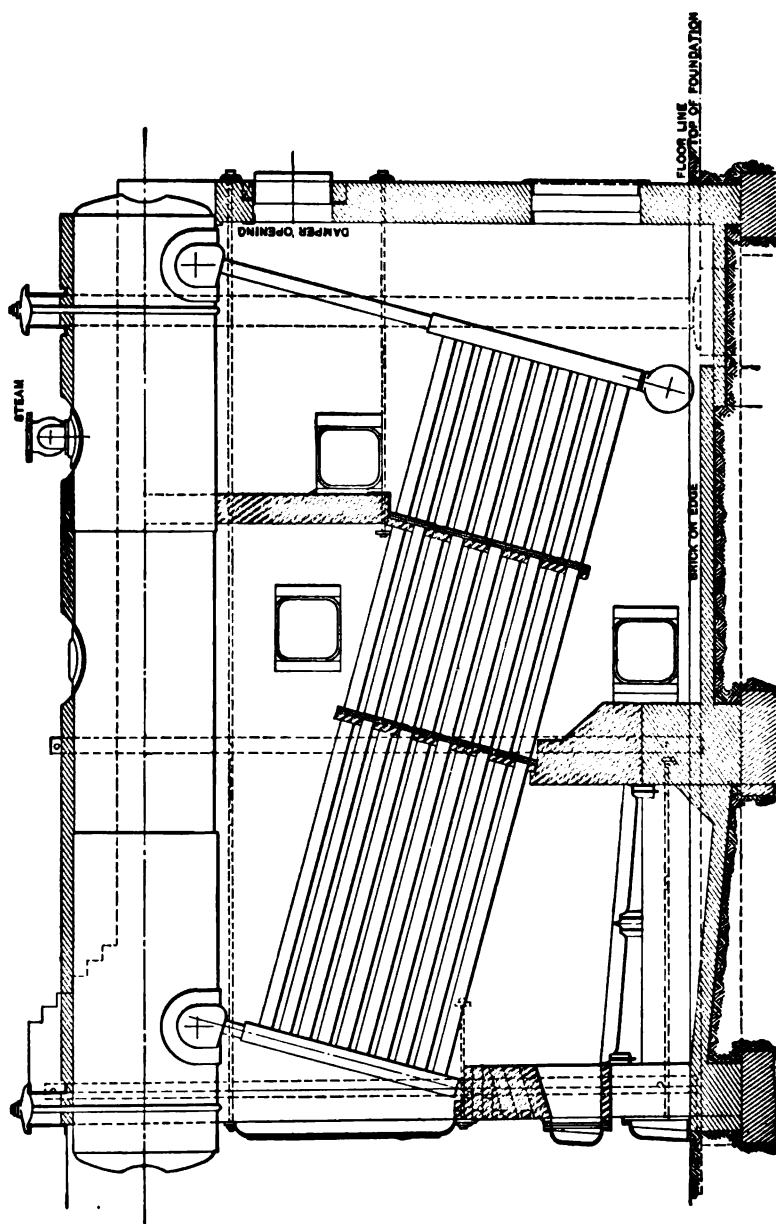


FIG. 29.—Babcock and Wilcox Boiler.

It is a difficult problem to make a selection between the two types. Where quick steaming and weight are prerequisite the water-tubular class is to be preferred, as also when very high pressures and hard forcing are necessary. When these requirements are lacking the choice becomes much more even, with a slight tendency to favor the fire-tubular class as being less complicated. Water-tubular boilers usually have a number of small accurately fitted parts, many of which are of special manufacture, a fact which is a disadvantage. Of the water-tubular class the straight-tube type are generally preferred, and practically tubes 2 inches diameter and larger are better than those smaller than 2 inches diameter.

For sake of illustration only a few examples need be described.

The Babcock and Wilcox Boiler (Figs. 29 and 29a) is one of the best known of its particular kind, is simple in design, and possesses strength, reliability and tightness. The tubes have expanded ends, and carry baffles to deflect the gases so as to reach all the heating surfaces.

The Stirling Boiler (Fig. 30) is of the bent-tube class and has proved very satisfactory. The tubes are difficult to examine and not especially easy to replace.

The Almy Boiler (Fig. 31) is chiefly used for small installations in marine work. It has been very successful, but is complicated by having many parts, necessitating a number of joints. It lacks facility for cleaning, although it has proved tight and durable.

The Niclausse Boiler (Fig. 32) is moderately simple, but requires accurate fitting. The boiler is not self-draining, but the parts are all accessible for cleaning and repairing.

The Belleville Boiler (Fig. 33) is of French design, and has become widely known through its marine use. It has not proved entirely satisfactory, probably due to the trouble to maintain it in good working order. It requires considerable skill in firing and handling, and the design is somewhat complex.

The Thornycroft Boiler (Fig. 34) is capable of withstanding a high degree of forcing. The bent tubes, however, make it difficult to clean and practically impossible to inspect internally.

The Yarrow Boiler (Fig. 35) is very simple and easy to clean and inspect. The tubes are difficult to replace, and their rigidity has been criticised. The tubes are generally small, about $1\frac{1}{4}$ inches to $1\frac{3}{4}$ inches diameter.

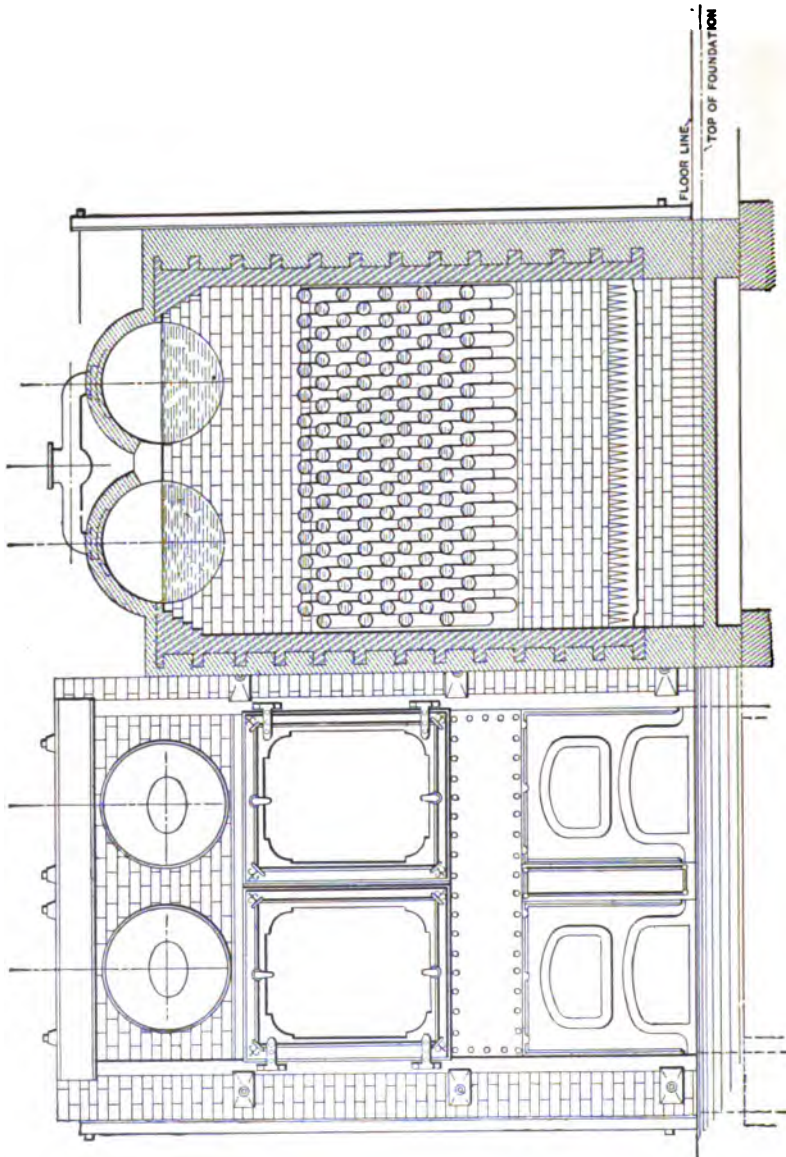


FIG. 28a.—Babcock and Wilcox Boiler. Front view and section of Fig. 29.

To Proportion a Boiler to Perform a Required Duty. Usually every engineer has his own ideas how to proceed to proportion a boiler or battery of boilers to meet certain requirements, but the various methods can be reduced to the three following schemes.

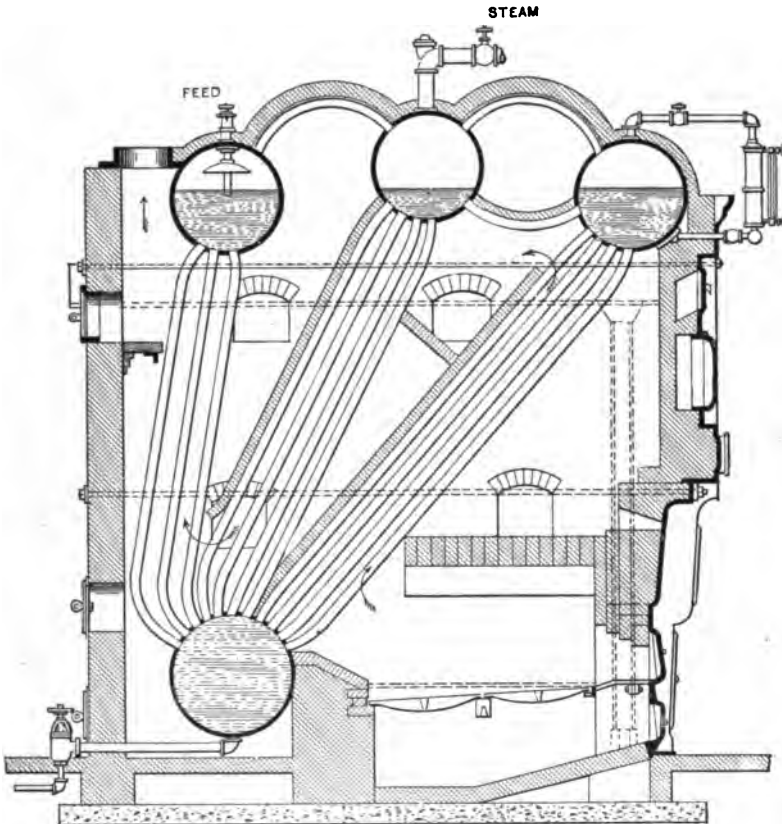


FIG. 30.—Stirling Boiler.

If any one be used, it is well to check the result by one of the others.

1. Determine the weight of water to be evaporated per hour. This should be the maximum and not the average weight, when variable loads are expected. This can be done by assuming, when the type of engine is known, the steam consumed per indicated horse-power hour and multiplying by the required horse-power.

For steam-heating plants it must be calculated from the radiating surface. (See works on that subject.) For industrial uses it must be obtained by experience. In all cases care must be exercised to see that all auxiliary engines and other users of steam are included.

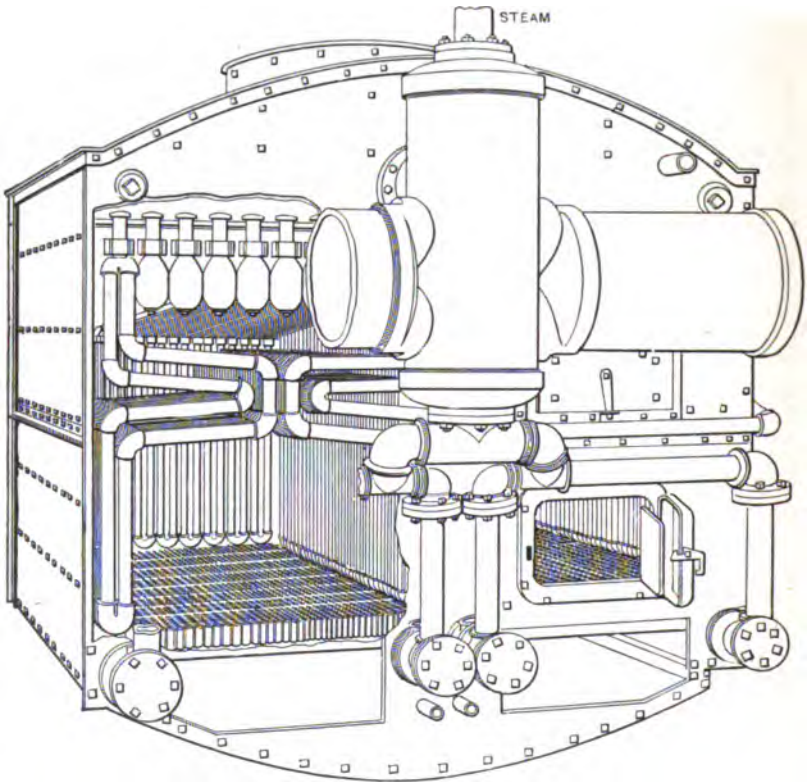


FIG. 31.—Almy Boiler.

Knowing the class of fuel, the evaporation per pound of fuel can be assumed, and dividing the first result by this rate of evaporation, the total weight of fuel can be determined. See that both total water evaporated and rate are either "actual" or "from and at 212°." Having thus determined the total fuel, the grate area can be calculated or assumed, and the height of chimney assumed or calculated for the required rate of combustion. Having deter-

mined the grate area, the heating surface can be proportioned from the principles stated under Heating Surface.

2. Knowing the class of engine to be used, assume the amount of heating surface per horse-power. This can only be done after

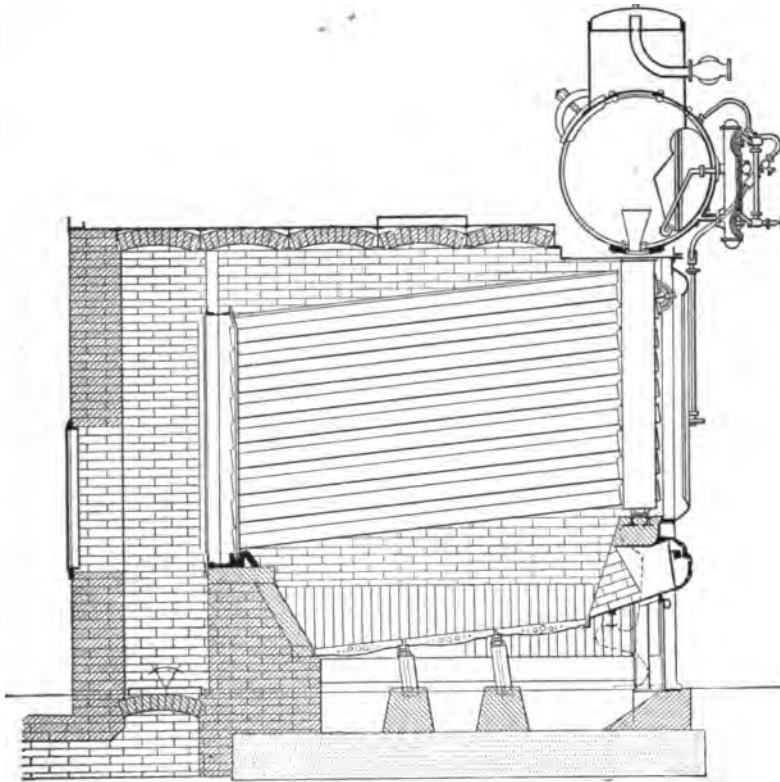


FIG. 32.—Niclausse Boiler.

considerable experience has been attained, or by comparison with some similar, successful plant.

Multiply by the total horse-power required and the product will be total heating surface. Remember that maximum and not average horse-power must be used, including the power of all auxiliary engines. The grate surface can be determined by the principles stated above.

3. Assume a coal consumption per horse-power per hour, and

then determine the total coal to be used by multiplying by maximum horse-power. Assume or calculate the rate of combustion

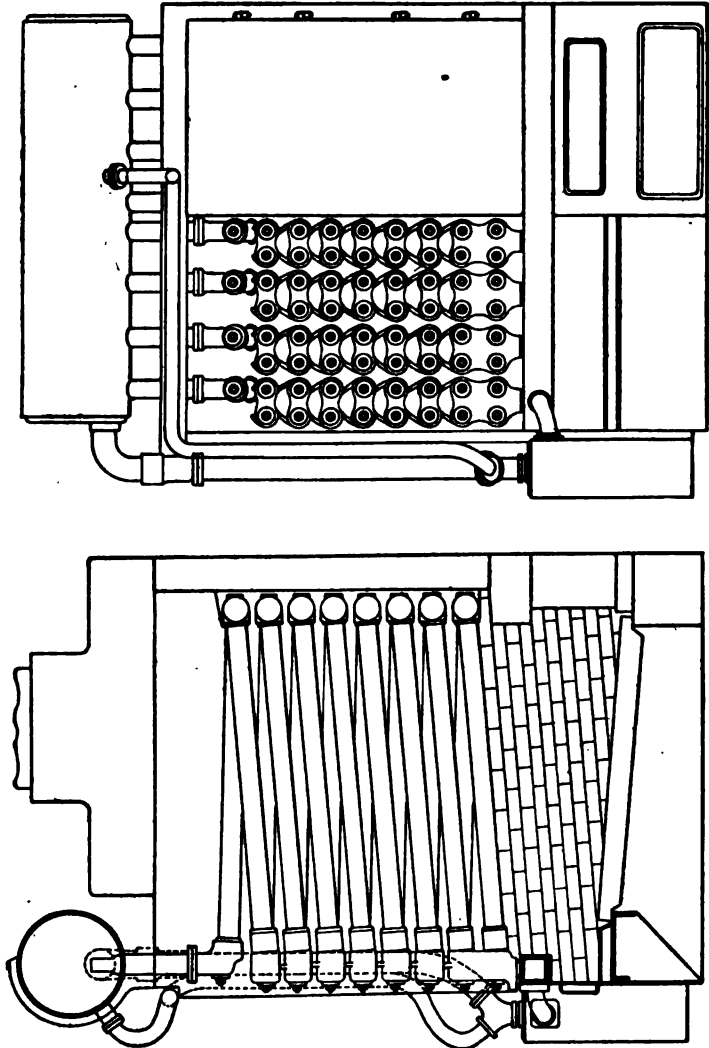


FIG. 33.—Belleville Boiler.

per square foot of grate per hour. Divide total amount of coal by this rate, and the quotient will be area of grate required. Then

proportion amount of heating surface as before, and make the stack high enough to give the rate of combustion.

The first method is generally the safest, but it is well to check the results.

After having fixed the general proportions, the number of boilers must then be determined. Remember that it is always

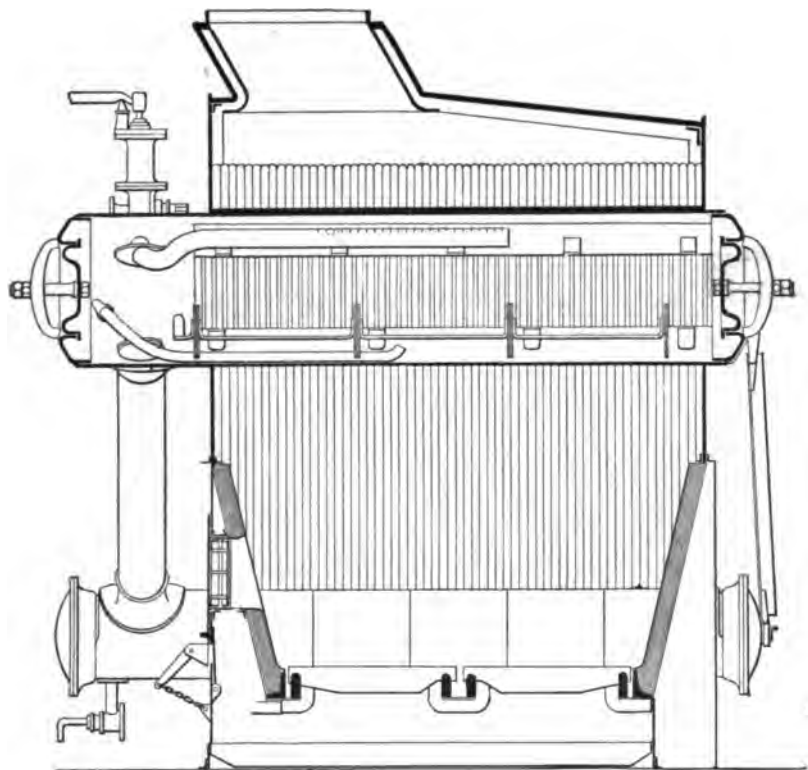


FIG. 34.—Thornycroft Boiler.

more economical to have ample boiler capacity, so as not to have to force beyond the normal rating.

It is wise and in most cases absolutely essential, especially when the plant is in steady operation, to have a surplus boiler, so that any one can be shut down for cleaning and repairs without affecting the plant.

While it is possible to calculate the size of boiler required by using the heat-units contained in the steam and in the coal, making allowance for the efficiency of the boiler, it is just as safe in practice to use the three methods given, as so many assumptions have to be made in every case.

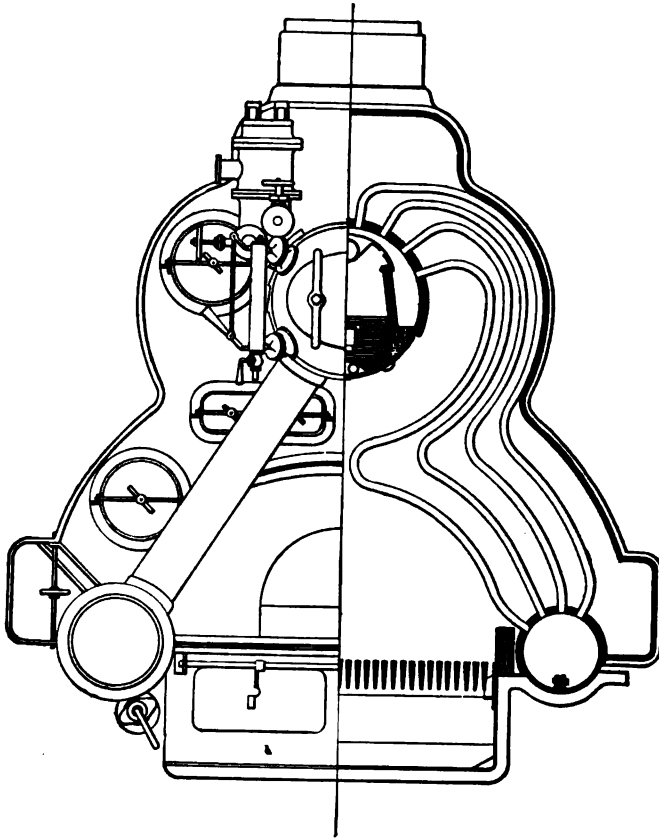


FIG. 34a.—Front view and section of Fig. 34.

Steam-space. The contents of a boiler are divided into water-space and steam-space. The former needs to be of only such capacity as safely to contain the water necessary for the generation of the steam. In water-tube boilers the water-space is very small

compared to many of the fire-tube boilers. In the latter the water-space has to be large, due to the design, so that there may be little danger of uncovering the highest heating surfaces.

The steam-space must vary in capacity due to the demand for steam; the smallest space being required for steady outflows of steam, and the largest for those that are intermittent.

For very intermittent flows, as the demands for steam by engines of long stroke making few turns per minute, the space is frequently

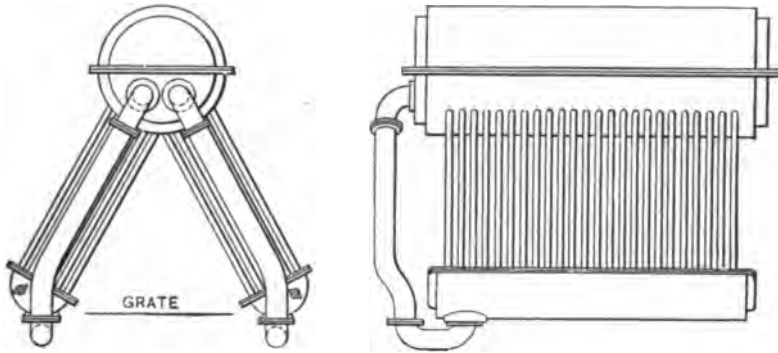


FIG. 35.—Yarrow Boiler.

figured in terms of the cylinder capacity. Thus for the walking-beam class of engine, as found on American river steamers, often having a single cylinder 50 inches diameter by 10 feet stroke, with steam at 30 to 50 pounds, the steam-space is about four times the cylinder capacity. Occasionally from 3 to $3\frac{1}{2}$ times has been found sufficient.

Table XIII. gives results based on practice, and is in the main taken from "Marine Engineering" by Seaton. It may be found that experience will modify the figures in estimating the required space for given conditions. The space does not depend on pressure, but on the volume of steam required. For uses other than for steam-engines the space can only be determined by experience.

With economical triple-expansion engines and quadruple-expansion engines the steam-space may be reduced 15 to 20 per cent.

When mechanical draft is used the space may be reduced 25 to 50 per cent, according to circumstances.

Priming. When ebullition is violent the bubbles of steam rise with such rapidity as to keep the water surface continually broken, and in consequence a considerable quantity of water in small

TABLE XIII
STEAM-SPACE; NATURAL DRAFT

General Type of Engine.	Per I. H. P. in Cubic Feet.
Small slow-running engines, steam-pumps, etc.	1.00 to 2.50
Moderately slow-running engines.	1.00
Faster-running engines.	1.00 to 0.80
Marine paddle-engines, direct-connected.	0.80
Marine walking-beam engines.	0.80 to 1.00
Fast-running stationary engines.	0.50 to 0.80
Marine engines, vertical direct-acting.	0.65
Naval and fast-running marine engines.	0.55 to 0.65

particles is carried up with the steam and retained in mechanical suspension. This action is called "priming" or "foaming." If there be no quiet place in the steam-space for this priming water to settle or separate from the steam, it is carried over into the steam-pipe and is liable to cause serious damage to the engine by knocking off the cylinder-heads or disarranging the valves. Unless super-heating or steam-heating surfaces be provided, nearly all boilers will pass off with the steam some priming or entrained water. Steam containing less than $2\frac{1}{2}$ per cent moisture may be said to be "commercially dry."

Priming or foaming may be caused by dirty or greasy water, oil in the boiler, etc., but it is often produced by the design, or by forcing a boiler beyond its proper capacity.*

When the steam-space is too small there is a fall in pressure at each efflux of steam, which will cause sudden and rapid ebullition. Priming from this cause can only be prevented by enlarging the steam-space, or by contracting the area of steam-pipe. This latter alternative may reduce the general efficiency of the engine by lowering the initial pressure, but must often be resorted to in order to save the great expense of condemning a boiler already built.

Priming is more often caused by the conformation of the boiler than by contracted steam-space. If the water-line in a shell be high, the flatness of the sides will cause priming, by tending to contract the effective water-separating surface, and by preventing a proper downward current without interfering with the upward current.

There are no special rules for area of water surface, as the value

* Boilers which often steam quietly with bad waters, sometimes foam when a change of water is introduced, probably due to the new water dissolving some of the deposit. Carbonate of sodium may cause foaming for a like reason.

of the area depends less on the quantity, by volume or weight, of steam generated than on the general design.

The design should provide for the following conditions in order to prevent priming:

First. Use of clean water. If the feed comes from a surface condenser, the oil from the cylinder lubricators should be separated or extracted.

Second. Sufficient steam-room to prevent fluctuation in pressure.

Third. As great a water surface as possible, so that the separation of steam may be least violent.

Fourth. The water surface should cover not less than one-half of the area of each space left for downward currents. This is most important.

Fifth. The steam-pipe should connect as high above the water-level as possible, and not directly over the hottest part. In boilers having comparatively small steam-space a collecting-pipe or dry-pipe, so placed on the inside and connected to the steam-pipe as to draw steam from all parts, will be found a good device. Such a pipe, frequently called an "anti-priming" pipe, should be stopped at the ends and have holes or slots on its upper side only. There should be a drain on under side. Baffle-plates may also be used inside the boiler, so arranged as to make it difficult for moisture to pass.

Sixth. The area of steam-pipe should not be made too large for the capacity of the boiler.

Some engineers proportion the water surface so that the velocity of steam rising from it shall not exceed $2\frac{1}{2}$ feet per second. If the velocity be greater, priming is almost sure to occur, as when once formed the water particles will have difficulty in settling back against a current of even less than half that velocity.

Thus, let V denote the cubic feet of steam generated per second, and S denote the minimum water disengaging surface in square feet. Then

$$S = \frac{V}{2.5}$$

In water-tubular boilers the water surface can be very materially reduced below the area required in fire-tubular boilers and still furnish dry steam.

Just how small the surface may be depends on the design, which is an all-important factor in this particular in all water-tubular boilers. The least amount can be determined only by experience.

In completing the design it will often be found that all the suggestions made in this chapter cannot be provided for. In such cases the design should be altered so as to retain the more important ones, and none of the suggestions should be discarded except after careful study and on the exercise of best judgment.

CHAPTER VI

CHIMNEY-DRAFT

Problem of Gravitation. Theory of Péclet as expressed by Rankine. Natural Draft. Rate of Combustion. Author's Experience. Area and Height of Chimney.

BEFORE any final calculations can be made for the general design the intensity of the draft must be considered, since upon it depends primarily the performance of a boiler. The quantity of fuel that can be burned is governed by the draft, which determines the rate at which the air-supply is drawn into the furnace. The quantity of air that will thus be drawn through the grate is chiefly controlled by the area of the flue, the height of the chimney and the temperature of the gases of combustion in excess of that of the outside air. Other things being equal, the velocity of the ascending current of hot gas may be taken as varying directly as the square root of the height of the chimney.

The problem of chimney-draft is really one of gravitation, and not of thermodynamics. No doubt heat must be supplied in order to maintain the velocity of the gases, but that same velocity is due not to the heat *per se*, but to the difference in weight of the hot gases and the cold outside air. For a full discussion of the subject, reference is made to papers in the Transactions of the American Society of Mechanical Engineers, Volume XI, 1890.

The best accepted theory is that of Péclet, which is admirably expressed by Rankine briefly as follows:

The draft of a furnace or the quantity of mixed gases that it will discharge in a given time may be estimated either by weight or by volume. It is often expressed by the pressure required to produce the current. This pressure is usually stated in "ounces per square inch" or in "inches of water."

In order to facilitate the discussion it may be assumed without serious error for all practical purposes, that the volume of the gases

of combustion at any temperature is equal to the volume of the air supplied when taken at the same temperature.

The volume at 32° F. of the gases may be assumed at 12.5 cubic feet for each pound of air supplied to the furnace.

Per Pound of Fuel,						Volume at 32° F. per Pound of Fuel,	
When 12 pounds of air are supplied.						150	cubic feet
"	18	"	"	"	"	225	" "
"	24	"	"	"	"	300	" "

The volume at any other temperature, such as T° , may be calculated from the formula

$$\text{Volume at } T^\circ = V = \text{Vol. at } 32^\circ \times \frac{T^\circ + 461^\circ.2}{493^\circ.2} = V_0 \frac{\tau}{\tau_0}.$$

The following results were obtained by this formula, and intermediate results may be interpolated with sufficient accuracy for practical work.

Temp. of Gases.	Supply of Air in Pounds per Pound Fuel.		
	12	18	24
	Volume of Gases per Pound Fuel in Cubic Feet.		
1112° F.	479	718	957
752°	369	553	738
572°	314	471	628
392°	259	389	519

Let w denote weight of fuel burned in furnace per second.

" V_0 " volume at 32° F. of air supplied per pound of fuel.

" τ_1 " absolute temperature of gas discharged.

" τ_0 " " " corresponding to 32° F.

" A " area of chimney in square feet.

" u " velocity of the current of gas in feet per second.

Then

$$u = \frac{w V_0 \tau_1}{A \tau_0},$$

which formula can be easily solved by interpolation from the list of values of $\frac{V_0 \tau}{\tau_0}$ just given, when the weight of coal burned is known or assumed.

Again, by transposition, the weight of fuel that may be completely burned can be calculated when the velocity of the current of hot gases is known or assumed, thus:

$$w = \frac{uA\tau_0}{V_0\tau_1}.$$

The weight of such a column of hot gas may be readily determined from the following approximate formula:

Density in pounds per cubic foot = $D = \frac{\tau_0}{\tau_1} \left(0.0807 + \frac{1}{V_0} \right)$,
in which 0.0807 is the weight of one cubic foot of air in pounds at 32° F.

Let l denote the length of chimney and flue leading to it, in feet.

“ m ” its hydraulic mean depth, or area divided by the perimeter.

“ f ” a coefficient of friction, which for gases over sooty surfaces is stated by Péclet to be about 0.012.

“ G ” a factor of resistance offered by the grate and bed of fuel to the passage of the air, which, according to Péclet, has a value of 12 when burning from 20 to 24 pounds of coal per square foot of grate per hour.

Then the “head” to produce the draft is

$$h = \frac{u^2}{2g} \left(1 + G + \frac{fl}{m} \right) = \frac{u^2}{2g} \left(13 + \frac{0.012l}{m} \right).$$

The head, h , is given in feet as the height of a column of hot gas, just sufficient to produce the unbalanced pressure which produces the current, u .

This head may be caused in three ways:

1. By the natural draft of the chimney;
2. By a jet or blast of steam or air;
3. By a fan or blowing-machine.

The head caused by the natural draft of the chimney is a column of hot gas of such a height as to have a weight equivalent to the difference between those of equal columns of cold air outside the chimney and of hot gases inside the chimney (Fig. 36).

It is more convenient to express the outside column of cold

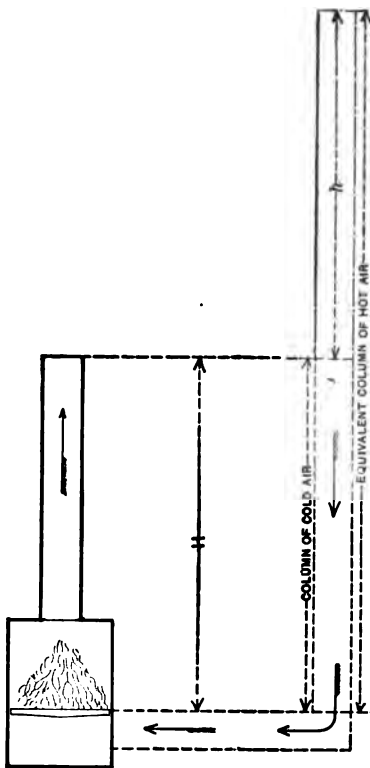


FIG. 36.—Chimney-draft.

air in feet of hot gas, that is, in the height of a column of hot gas having the same weight as the column of cold air. This is done by computing the weight of a column of cold air one square foot in section and as high as the top of the chimney is vertically above the grate, and dividing the result by the weight of a cubic foot of hot gas. If from this result the height of the chimney above the grate be subtracted, which is the height of the hot-gas column inside the chimney, the difference will be the "head," h , which causes the draft. Thus

Let H denote the height of the chimney above the grate.

Let τ_2 denote the absolute temperature of the external air.

Then

$$h = \frac{H \frac{\tau_0}{\tau_2} (0.0807)}{\frac{\tau_0}{\tau_1} \left(0.0807 + \frac{1}{V_0} \right)} - H = H \left(0.96 \frac{\tau_1}{\tau_2} - 1 \right),$$

or

$$H = \frac{h}{\left(0.96 \frac{\tau_1}{\tau_2} - 1 \right)}.$$

From this last equation can be calculated the height of chimney to produce a given draft.

It is evident that, for a given outside temperature, there must be some other temperature for the hot gases which will produce the

"best" draft, that is, the draft which will discharge the maximum weight of gases in a given time. The velocity of the current, u , or the strength of the draft, will increase with the values of τ_1 , but, owing to the rarification of the hot gases, the maximum discharge in weight will be at some fixed temperature for every value of τ_2 .

Since the velocity of the current of hot gases is proportional to \sqrt{h} , it must also be proportional to $\sqrt{(0.96\tau_1 - \tau_2)}$. The density of the hot gas is proportional to $\frac{1}{\tau_1}$. Also, the weight discharged per second is proportional to the velocity times density, or to $\frac{\tau_1}{\sqrt{0.96\tau_1 - \tau_2}}$.

This expression becomes a maximum when

$$\tau_1 = \frac{2\tau_2}{0.96} = 2\frac{1}{12}\tau_2.$$

Therefore the greatest weight of hot gas is discharged from the chimney when the absolute temperature of the hot gas in the chimney is to that of the external air as 25 is to 12.

When this condition is fulfilled, $h = H$. That is, for the "best" chimney-draft, or condition for maximum discharge of gases as measured by weight, the head when expressed in the height of a column of hot gas is equal to the height of the chimney.

Since the external air may be taken as having an average temperature of 50° F., equivalent to 511°.2 absolute, and since the corresponding temperature of chimney-gases for maximum discharge would be 1065° absolute, corresponding to 603°.8 F., it may be stated that the "best" draft to create a maximum discharge will be produced when the temperature of the gases in the chimney is nearly sufficient to melt lead.

2. When the draft is produced by a jet or blast in the chimney, an artificial current is caused by the impact of this jet against the hot gases. The head is equivalent to that atmospheric head, or natural head, which would be required to produce the same velocity.

3. When the draft is produced by a fan or blowing-machine, the effect is that an artificial current is caused. The head, as in the

previous cases, is due to the unbalanced pressures between the external air and internal gases.

The conditions of artificial draft will be considered in Chapter XI.

Natural Draft. Experience with many chimneys of various sizes has given proof of the general correctness of the analysis of Rankine and Péclet. There have been cases reported in which results differ, but such have shown, on careful inspection, either an incorrect application of the formulæ or the introduction of conditions not properly accounted for. The chief difficulty in using the Péclet formulæ is the selection of proper values for the constants.

In practical application, for the determination of the height of chimney-stack, values for w , u , and A must be first settled. A value for w is determined by the conditions of the new plant, since a boiler or battery of boilers is designed to evaporate a given amount of water. This would necessitate the combustion of a certain amount of coal per second. The value of A is generally made about one-seventh, one-eighth, or one-ninth the area of the grate, as such proportions have given good results. The larger values are for bituminous coal or for short stacks, while the smaller are for anthracite or for tall stacks. When w and A are assumed, the value of u can be calculated. If a value be assumed for u , then w can be calculated.

Without serious error for preliminary work, the expression $\frac{fl}{m}$ can be taken as unity, and a value for h can be easily determined. The value of H , which was to be found, then depends on the assumed temperature of the gases which may be expected in the proposed plant.

The frictional resistance, f , must necessarily be a variable quantity, but probably does not differ much in ordinary cases from the value assigned by Péclet. The resistance to draft, offered by the grate and fuel, G , is necessarily large, requiring the major part of the head. It is a loss that is an essential part of the system. While the value of Péclet was determined from experiments where 20 to 24 pounds of coal were burned per square foot of grate per hour, the real value may be somewhat less for lower rates of combustion. Some engineers use 11 for rates of 15 pounds. The true value must depend not only on the rate, but on the thickness of fire, size of coal and kind of grate.

From what has been stated it will be seen that in practical application many assumptions have to be made under conditions which are more or less rapidly changing. In consequence engineers are prone to adopt empirical rules for the determination of their results, and experience has proven such methods sufficiently accurate for all practical cases.

Furthermore, attempted accurate calculations are rendered really approximate due to effects of location of chimney, for which no exact allowance can be made. The draft will be stronger in chimneys built on high ground than in those in valleys; or in those on open plains than in those surrounded by high buildings. Wind also may help the draft considerably, especially if the chimney-top be properly designed. For these reasons the funnels of steamers always have a better draft than those of equal height on land.

Rate of Combustion. The rate of combustion primarily depends on the strength of the draft, which will vary approximately, under similar conditions, as the square root of the height of chimney. It will also depend upon the grade or quality of fuel, its dryness, character of grate, proportions of combustion-chamber, amount of air supplied and its initial temperature.

The rate as generally stated in text-books for various kinds of boilers is given in Table XIV. See Rankine's "Steam-engine."

TABLE XIV
RATES OF COMBUSTION; NATURAL DRAFT

Kind of Boiler.	Pounds of Coal per Square Foot of Grate Surface per Hour.
Cornish boilers, slowest rate.....	4
" " average rate.....	10
Factory boilers, average rate.....	12 to 16
Marine boilers, average rate.....	16 to 24
Ordinary boilers, quickest rate with dry coal, air admitted under grate only.....	20 to 23
Ordinary boilers, quickest rate with soft coal, with area of air- holes above grate equal to $\frac{1}{4}$ area of grate.....	24 to 27

Such results were based on the practice of the time and with the usual height of stack as then obtained.

The actual rate of combustion will depend on the conditions, however, stated above.

The anthracites require a stronger draft than the bituminous coals, therefore with equal draft-strengths the hard coals burn at a lower rate than the soft ones.

Assuming ordinary factory conditions, the average coals of the various qualities may be taken relatively as in Table XV. It must be remembered that some of the soft coals will burn still more freely than here indicated, while some of the anthracites less rapidly. The figures must be treated as averages.

TABLE XV
RELATIVE RATES OF COMBUSTION FOR COALS

Kind of Coal.	Weight of Coal Burned per Square Foot of Grate.	Area of Grate per Pound Consumed.
Good anthracites.	1.00	1.00
Good semi-anthracites and bituminous.	1.15	0.90
Ordinary low grades, soft.	1.50	0.70
“ “ “ hard.	0.90	1.10

Restricted draft area will reduce the rate of combustion, and too great an area, within moderate limits, does not increase the rate as rapidly as it lowers the efficiency of the boiler. The area to give best results is entirely dependent on the character of the coal and on the strength of the draft. For ordinary, average conditions the area should be about one-eighth the area of grate.

Rate Depends on Quality and Size of Fuel. All conditions of chimney, etc., being the same, the rate of combustion will vary; wood will burn the fastest, then bituminous coal, then semi-bituminous, semi-anthracite and finally anthracite. The small sizes of anthracite will burn more slowly than the larger ones. Consequently for equal rates of combustion the highest chimney is required for the small sizes of anthracite, and the lowest for bituminous coals and wood.

The rate is found to vary for the different fuels with the type and setting of the boilers, those offering the greatest resistance to draft will have the lowest rate. As the variation for the different qualities of fuel is so irregular, no definite relation can be expressed and much depends on experience.

Walter S. Hutton, in "Steam-boiler Construction," states rates of combustion and draft-pressures for various chimney heights, but

does not state the exact conditions. Table XVI gives his results, as printed in "Mechanical Draft," Sturtevant and Co. The same table gives results obtained by Prof. W. P. Trowbridge based on uniform data for all, but without allowing for variation in kind or quality of fuel. See "Heat and Heat-engines," by Trowbridge.

TABLE XVI
RATES OF COMBUSTION FOR DIFFERENT CHIMNEY HEIGHTS

Height of Chimney above Grate, in Feet.	Pounds of Coal per Square Foot of Grate per Hour.		Height of Chimney above Grate, in Feet.	Pounds of Coal per Square Foot of Grate per Hour.	
	Hutton.	Trowbridge.		Hutton.	Trowbridge.
25	10	8.5	100	22	19.0
50	16	13.1	110	24	20.0
60	17	14.5	120	27	
70	18	15.8	150	40	
80	19	16.9	200	60	
90	20	18.0	250	80	

The formula of William Kent (Trans. Am. Soc. Mechanical Engineers, Vol. VI), based on observation, is:

$$\text{Rate} = 2.1\sqrt{H},$$

in which H denotes the height of chimney, the area of the chimney being taken at one-eighth of the grate. For any ratio of grate to area of chimney the formula becomes

$$F = \frac{A\sqrt{H}}{0.06},$$

in which A denotes area of chimney; F , the pounds of coal burned per hour.

Prof. R. H. Thurston states the following formula:

Under best conditions, as in marine work, with anthracite coal,

$$\text{Rate} = 2\sqrt{H} - 1.$$

Under more ordinary conditions, as in general stationary work, with anthracite coal,

$$\text{Rate} = 1.6\sqrt{H} - 1.$$

Best Welsh and Maryland semi-anthracite or good bituminous coals should give

$$\text{Rate} = 2.3\sqrt{H} - 1.$$

The less valuable soft coals with air admitted above the grate, in proportion of area of holes to grate as 1 to 36, should give

$$\text{Rate} = 3\sqrt{H} - 1.$$

Under ordinary conditions of stationary practice each 10 feet additional height to the chimney will increase the draft about one-twelfth of an inch water-pressure, and each half-inch of pressure will increase the rate of combustion about 10 pounds of coal per hour. George W. Melville, U. S. N., states (Transactions of Marine Congress, Chicago, 1894) that, based on naval experience, each additional 10 feet in height of funnel increased the pressure of draft about one-eighth inch, and that a 100-foot funnel would give a rate of about 25 pounds of coal per square foot of grate per hour.

Author's Experience.—From observations made by the author, the figures given by Hutton represent rates for free-burning bituminous coals, and those of Trowbridge for anthracites, while both are only obtained in stationary work when everything is favorable for rapid combustion, open dampers and direct connections to stacks without tortuous passages for the gases through the boiler. For similar reasons, the empirical formulæ are apt to furnish higher values than will be obtained in daily practice. Possibly the formulæ and the observations were based on experiments, where the escaping gases were at a greater temperature than is usually the endeavor under present practice to secure. The temperature of the gases should be as low as is consistent with the requirements of draft, and the draft should be so regulated by proper design of chimney area and height as to permit of the lowest possible temperature.

In a new design it is always best to have ample boiler power, as losses are incurred when boilers are forced to produce the required steam, through the continual opening of the fire-doors and the raking of the fires, thus cooling the combustion-chamber and sifting good coal into the ash-pit. Ample boiler capacity can be secured by assuming a low rate of combustion in proportion to the height of chimney, but a low rate of combustion has not been found economical, and much must depend on personal experience.

Area and Height of Chimney.—The area and height of chimney are closely allied, since the product of area times velocity measures the quantity of gases discharged, and the velocity is dependent upon the height. In general, the shorter the chimney the larger should be the area, and conversely.

An empirical formula, largely used by engineers, is that of William Kent, explained in Transactions Am. Soc. M. E., Vol. VI, namely:

Let E denote effective area of chimney.

“ A “ actual area of chimney.

“ H “ height of chimney.

“ $H.P.$ “ the horse-power, based on a coal consumption of 5 pounds of coal per hour. If the expected rate of coal per horse-power be different, then the result must be corrected by multiplying by the ratio of 5 to the maximum expected rate of consumption per horse-power.

Then

$$E = A - 0.6\sqrt{A};$$

$$H.P. = 3.33E\sqrt{H} = 3.33(A - 0.6\sqrt{A})\sqrt{H};$$

$$E = \frac{0.3H.P.}{\sqrt{H}}.$$

This formula is based on the assumptions that the draft varies as the square root of the height, that the retarding of the ascending gases by friction may be considered as equivalent to a diminution of the area of chimney equal to that of a layer 2 inches in thickness, and that the power varies directly as this effective area.

Then, for round flues, diameter = diameter of $E + 4$ inches;

“ square “ side = square root of $E + 4$ inches.

The Riter-Conley Manufacturing Company* have adopted a formula for the horse-power of chimneys which is simpler and appears to give better results than Kent's, namely,

$$H.P. = \frac{5D^2\sqrt{H}}{2},$$

* Of Pittsburg, Pa. Kindness of Mr. Wm. C. Coffin, Vice-President.

in which *H.P.* is the same as Kent's definition, *D* the internal diameter in feet, and *H* the height above grate in feet.

John W. Hill states a formula as follows:

$$A = \frac{1.8 \text{ grate surface}}{\sqrt{H}},$$

where *A* denotes the area of flue and *H* the height above the grate. This formula is based on experience with Western bituminous coals under natural draft, and burning at rates varying from 15 to 25 pounds per square foot of grate per hour.

Prof. A. C. Smith states a formula in which *F* denotes the coal burned per hour on the grate in pounds, thus:

$$A = \frac{0.0825F}{\sqrt{H}},$$

$$H = \left(\frac{0.0825F}{A} \right)^2.$$

Many engineers simply adopt the following proportions, the grate surface being taken as unity. For coals midway between anthracite and bituminous use intermediate values. For rapid rates of combustion use proportionately increased values.

	Bituminous.	Anthracite.
Area over bridge wall.	$\frac{1}{5}$	$\frac{1}{7}$
Area through tubes, or calorimeter. . .	$\frac{1}{6}$	$\frac{1}{8}$
Area of chimney-flue.	$\frac{1}{7}$	$\frac{1}{9}$

It is well to proportion an ample chimney for all cases, and to use the damper if the draft be too strong. There is then a reserve of power for use if necessary.

Chimneys for very large powers need not have so great an area as those for small power, in proportion to their size. There are cases, with large batteries of boilers, where the stack has worked well when the flue-area is less than $\frac{1}{20}$ of grate surface.

For large batteries of boilers it is often found better to use a number of smaller chimneys than a single large one, as so much resistance is offered by the long connections or breechings, as to

require the single chimney to be extra high. Furthermore, the boilers nearest the chimney will rob those farther away.

As very high chimneys are expensive to construct, drafts produced by mechanical means are becoming more general, and can often be operated and maintained for the interest on the saving of cost of stack. Under conditions of artificial draft the areas mentioned just above should be increased to accommodate the greater quantity of fuel burned.

In marine practice it is customary to increase the areas above those adopted for stationary practice. The area of stack is often determined by the general appearance of the vessel, although the modern tendency is toward high funnels with somewhat reduced diameters.

For equal heights the draft is stronger in marine boilers than in similar ones on land, so that the area in any event should be greater.

CHAPTER VII

MATERIALS

Cast Iron. Wrought Iron. Rivet-iron. Charcoal-iron for Boiler-tubes. Wrought Steel. U. S. Naval Requirements for Boiler-steel. Steel Rivets. Steel for Boiler-braces. Mild Steel Affected by Temperature. Cast Steel. Copper. Brass. Bronze. Muntz's Metal.

THE materials used in boiler-making are chiefly cast iron, wrought iron, steel, cast steel, copper and brass.

Cast Iron is used for many of the boiler fittings, supports and accessories. It has gradually been rejected as a material to be worked into parts subject to great variation of heat and pressure, on account of its brittle and uncertain nature when placed under tensile stress. If carefully worked and finally cast only after repeated remeltings, so as to render it homogeneous, it has shown remarkable properties for endurance, even under severe conditions. Owing to its lack of tensile strength, parts of cast iron have to be made very thick, and are thus liable to internal stresses not visible from external examination.

Cast iron is largely used for furnace and ash-pit doors (although many are now being made of steel and cast steel), valve-casings, crosses and tees for boiler-mountings, grate-bars and bearers, man-hole and handhole covers, supports, supporting lugs, boiler-fronts, etc.

Steam-pipes are occasionally made of cast iron, but this practice is not to be recommended. When so used these pipes should be carefully drained, so as to prevent any water collecting in them from condensed steam. Such water must be blown out before the full steam-pressure be turned on the piping.

Many of the more important pieces are best made of malleable cast iron; and this material is largely used in the construction of some parts of water-tubular boilers, as it has a ductility from four to six times greater than ordinary cast iron.

When cast-iron fittings are mounted on boilers, use care not to permit water to lodge in them. Many boilers have been exploded by permitting water to collect on top of cast-iron stop-valves. This can be best prevented by designing a proper placing of the valve or mounting so as to be self-draining, or by fitting a drain-pipe which must be opened first.

The cast irons are frequently known by numbers; thus number one contains the most graphite or carbon, while the higher numbers the least. The numbers run from one to seven. They are frequently known by names of the districts in which made. The commonest designation divides the irons into three classes, thus: number one, gray iron or foundry iron; number two, mottled iron; and number three, white iron. The numbers two and three are sometimes called "forge" irons.

The best irons for strong castings are made from mixtures, thus combining desired qualities as strength, fluidity, close-grained, etc.

Hardness can be obtained by mixing steel scrap, which will give a close-grained and strong casting. The scrap may be added in amounts as much as 10 to 15 per cent. Little is gained by increasing the amount of scrap, as the iron becomes so hard as to be available only for thick castings without complications of form.

The strongest cast irons are of a light-gray color, and on fracture should exhibit a close, uniform grain. The tensile strength should not be less than 18,000 pounds, and the crushing strength 110,000 pounds per square inch. For all important work the iron should be made from selected scrap, or should be remelted at least once. Remelting improves the quality until the process has been repeated about twelve times.

Wrought Iron is now little used, having yielded its place to wrought steel. The chief objection to wrought iron is its lack of uniformity or homogeneity, due to the existence of blisters and laminations, often difficult to detect from surface examination. These defects can usually be detected by tapping the sheet with a light hammer, especially when the sheet is supported on two diagonally opposite corners.

Many engineers prefer wrought iron to steel plates for tank and other cheap work, since the poorest grades of the latter are likely to be used, and such steel will not stand the same amount of rough usage in handling and bending as the iron.

Wrought iron is never obtained commercially pure, but is always mixed with more or less slag, and such elements as carbon, sulphur, phosphorus, silicon and manganese.

The usual proportion of these elements should not exceed the following amounts, expressed in percentage:

	From	To
Carbon.	0.020	0.20
Sulphur.	0.000	0.01
Phosphorus.	0.050	0.25
Silicon.	0.050	0.30
Manganese.	0.005	0.05

Sulphur in excess of the above quantity makes the iron brittle when heated to redness, or red-short; phosphorus, brittle when cold, or cold-short. The effect of silicon and manganese appears to be important.

Good iron boiler-plate should show a tensile strength of from 48,000 to 56,000 pounds per square inch, an elongation of not less than 10 or 15 per cent in test pieces 8 inches between marks, and a reduction of area of not less than 40 per cent. The elastic limit is about 27,000 pounds.

The appearance of the fracture depends on the manner in which the specimen was broken. When the pressure is applied slowly, good wrought iron should exhibit a fibrous and irregular fracture, while a poorer quality will show a more regular fracture, the fibres being broken across and mixed with crystalline structure. The fineness and coarseness may be taken as a test of quality. On the other hand, when suddenly torn apart or broken by a sharp transverse blow, good iron will show a fine crystalline fracture, while the poorer qualities exhibit a much coarser structure. The fibrous nature may all be lost by the sudden action of the fracture. Wrought iron should never be judged as to quality, unless the method of making the fracture be positively known.

Wrought iron is preferred by many for rivets, although recent improvements in steel manufacture have been instrumental in introducing the latter in preference to iron, especially for heavy work. As iron is less apt to be injured by working cold and also by being overheated in the forge, iron rivets should be used when driven cold or for use in cheap work.

Rivet-iron should be soft and of the best quality, as it has to withstand hard usage. The tensile strength should be not less than 50,000 pounds per square inch, and its resistance to shearing is usually from 38,000 to 40,000 pounds per square inch. A rivet should bend double when cold, without fracture; and when hot, the head should be capable of being flattened to about $\frac{1}{8}$ inch in thickness without fraying at the edge. The minimum elongation in specimens 8 inches long should be not less than 18 per cent, and the minimum contraction should be not less than 40 per cent. When the bar is bent after having been nicked on one side, the fracture should be fibrous.

Tests should not be made on bars less than 8 inches long between marks, nor of less sectional area than $\frac{1}{4}$ square inch.

Stays and all pieces that have to be forged or welded are best made of iron, of the same quality as for rivets.

Wrought iron is used for making boiler-tubes, although many are made of steel. Charcoal-iron is considered the best grade for tubes. It is very difficult to detect the difference between charcoal-iron and steel tubes by surface inspection, but the material can be readily identified from the fracture.

Wrought iron for tubes should have a tensile strength of not less than 49,000 pounds and an elongation of at least 15 per cent in a length of 8 inches.

The requirements of the Bureau of Steam Engineering, U. S. Navy Department, 1897, for charcoal-iron boiler-tubes are as follows:

SPECIFICATIONS FOR CHARCOAL-IRON BOILER-TUBES.

1. The tubes must be made of the best quality of knobbled, hammered charcoal-iron, and tests will be made both of the skelp and of the finished tubes to determine whether or not such material has been used. Pig iron only will be used in the manufacture of the blooms. The tubes must be of uniform gauge throughout, and be smooth and free from rust, scale, pits, laminations, or imperfect welds.

2. Strips one-half inch in width will be cut lengthwise from the tubes; these strips will be heated to a bright cherry-red, in daylight, and plunged (while at this heat) in water at a temperature of 80° F.; after being so quenched a strip must bend, cold, back upon itself without crack or flaw. Strips so heated and quenched must not show lamination when hammered.

3. The end of a tube will be heated to a bright cherry-red, in daylight, for a distance equal to one and one-half ($1\frac{1}{2}$) times its diameter, and a taper-pin, at a blue or dull-red heat, driven in. The tube must stretch to one and one-eighth ($1\frac{1}{8}$) times its original diameter without split or crack. The taper of the pin must not exceed one in eight ($1\frac{1}{8}$ inches to the foot), and its surface must be smooth.

4. Each tube will be subjected to an internal hydrostatic pressure of 500 pounds to the square inch.

5. These tests will be made by an engineer officer, and one tube will be selected by him, personally, for test from each lot of 250 or fraction thereof, unless in his opinion a larger number must be tested to enable him to form a proper estimate of the quality of the whole lot. The surface inspection will, of course, be made on each tube of the whole lot.

6. Failure to pass satisfactorily any of the above tests will be cause for rejection of the whole lot of tubes.

Steel. The material almost exclusively used for boiler construction at the present time is mild steel, since it is stronger, has a higher elastic limit, greater elasticity, and is more homogeneous in structure than wrought iron.

Steel may be considered as an alloy of iron, and can be placed midway between the cast and wrought irons.

Only the milder steels, those containing low percentages of carbon, are used for boiler-work in order to take advantage of the greater homogeneity and higher ductility.

The best boiler-steel plates are made by the Siemens-Martin or open-hearth process, although many of the cheaper boilers are made of Bessemer plates.

Steel is liable to injury while working, and all pieces that have been subjected to a partial heat, as in flanging, should be annealed to restore the original toughness. Steel is liable to become brittle when worked cold. The act of punching for rivet-holes injures steel plates, especially if they are hard. Steel plates should always be drilled, or if punched, the holes should be reamed out to remove the injured material.

Steel is used for rivets and tubes, and good results are obtained if care is taken in the selection of the material.

The tensile strength is usually limited between 60,000 and 73,000 pounds per square inch. If the limits are placed close, the cost rapidly increases. The object of having a maximum as well as a minimum limit is to give uniformity to all parts alike, since the

steel becomes hard and loses ductility and elasticity as the strength increases. For all flanged pieces the tensile strength is generally limited between 52,000 and 60,000 pounds.

For ordinary commercial work the elongation in specimens 8 inches long should be at least 20 per cent for shells, 25 per cent for flanged pieces, and 27 per cent for stays; and the contraction of area, 50 per cent for plates $\frac{1}{2}$ inch thick and less, 45 per cent for $\frac{1}{2}$ to $\frac{3}{4}$ inch, and 40 per cent for all over $\frac{3}{4}$ inch.

All the boiler accessories not directly contributing to strength, such as fire-doors, ash-pit doors, smoke-pipes, etc., may be made of a poorer grade of steel.

The requirements of the U. S. Navy Department, 1899, are as below:

SPECIFICATIONS FOR BOILER-PLATE AND SHAPES.

BOILER-PLATE. (CLASS A.)

Kind of Material.—Steel for boiler-plates shall be made by the open-hearth process, and not show more than thirty-five one-thousandths of 1 per cent of phosphorus nor more than three one-hundredths of 1 per cent of sulphur, and be of the best composition in other respects.

Test Specimens.—One tensile-test piece, taken longitudinally, and one bending-test piece, taken transversely, shall be cut from each plate as rolled for boilers, as directed by the Naval Inspector. The cold-bending pieces must not have their sheared nor planed sides rounded off, but the sharpness of the edges may be taken off with a fine file.

Tensile Tests of Class A No. 1 Plates.—Test specimens must show a tensile strength of at least 74,000 pounds per square inch, with an elastic limit of 40,000 pounds per square inch, and an elongation of at least 21 per cent in eight (8) inches.

Tensile Tests of Class A No. 2 Plates.—Test specimens must show a tensile strength of at least 66,000 pounds per square inch, with an elastic limit of at least 36,000 pounds per square inch, and an elongation of at least 23 per cent in eight (8) inches.

Tensile Tests of Class A No. 3 Plates.—Test specimens must show a tensile strength of at least 60,000 pounds per square inch, with an elastic limit of at least 32,000 pounds per square inch, and an elongation of at least 25 per cent in eight (8) inches.

Cold-bending Test.—One specimen cut from each Class A No. 1 and Class A No. 2 plate, as finished at the rolls for cold-bending test, shall bend round a curve the diameter of which is equal to the thickness of the plate tested till the sides of the specimen are parallel, without signs of fracture on the outside of the bent portion.

Cold-bending Test.—One specimen cut from each Class A No. 3 plate shall bend flat on itself without signs of fracture on the outside of the bent portion.

Inspection for Surface and Other Defects.—Plates must be free from slag, foreign substances, brittleness, laminations, hard spots, injurious sand or scale marks, scabs, snakes, and injurious defects generally.

Shearing.—Boiler-plates thirteen-sixteenths ($\frac{13}{16}$) of an inch thick and over shall not be sheared closer to finished dimensions than once the thickness of the plate along each end, and one half the thickness of the plate along each side. This allowance shall be made by the contractor on his order, and the manufacturer shall shear to ordered dimensions.

Weight and Gauge.—Contractors shall enter on their orders to manufacturers both weight per square foot and gauge of plates. Plates shall not vary from the specified weight more than the following amounts: 2 per cent below and 7 per cent above for plates more than one hundred and ten (110) inches wide; 2 per cent below and 6 per cent above for plates between one hundred (100) and one hundred and ten (110) inches wide; 2 per cent below and 5 per cent above for plates between eighty (80) and one hundred (100) inches wide; 2 per cent below and 4 per cent above for plates between sixty (60) and eighty (80) inches wide; and 2 per cent below and 3 per cent above for plates less than sixty (60) inches wide.

Gauge.—No plate must, at any point, fall below the specified thickness more than the following amounts: Six one-hundredths ($\frac{6}{100}$) of an inch for plates more than one hundred and ten (110) inches wide; five one-hundredths ($\frac{5}{100}$) of an inch for plates between one hundred (100) and one hundred and ten (110) inches wide; four one-hundredths ($\frac{4}{100}$) of an inch for plates between eighty (80) and one hundred (100) inches wide; three one-hundredths ($\frac{3}{100}$) of an inch for plates between sixty (60) and eighty (80) inches wide; and two one-hundredths ($\frac{2}{100}$) of an inch for plates less than sixty (60) inches wide.

Oil-tempering and Annealing.—It is left optional with the manufacturers to oil-temper and anneal Class A No. 1 plates of a thickness greater than one (1) inch, in order to get the requirements of these specifications; but the oil-tempering and annealing must be done before the plate is submitted to the Naval Inspector for tests.

SPECIFICATIONS FOR RODS, SHAPES, AND FORGINGS FOR BOILER BRACING.

BOILER BRACING. (CLASS A.)

Kind of Material.—Steel for stay-bolts and braces shall be made by the open-hearth process, shall not show more than thirty-five one-

thousandths of 1 per cent of phosphorus, nor more than three one-hundredths of 1 per cent of sulphur, and shall be of the best composition in other respects. The drillings for chemical analysis shall be taken from the same objects as the tensile-test pieces.

One ton of material for boiler braces, from the same heat, shall constitute a lot from which two tensile and one cold-bending test specimens shall be taken, each from a different object.

Treatment.—All material for boiler bracing shall be annealed as a final process.

Tensile Tests of Class A No. 1 Bracing.—Test specimens must show a tensile strength of at least 74,000 pounds per square inch, with an elastic limit of at least 40,000 pounds per square inch, and an elongation of at least 22 per cent in eight (8) inches, or 26 per cent in two (2) inches, in case 8-inch specimens cannot be secured.

Tensile Tests of Class A No. 2 Bracing.—Test specimens must show a tensile strength of at least 66,000 pounds per square inch, with an elastic limit of at least 36,000 pounds per square inch, and an elongation of at least 24 per cent in eight (8) inches, or 28 per cent in two (2) inches, in case 8-inch specimens cannot be secured.

Tensile Tests of Class A No. 3 Bracing.—Test specimens must show a tensile strength of at least 60,000 pounds per square inch, with an elastic limit of at least 32,000 pounds per square inch, and an elongation of at least 26 per cent in eight (8) inches, or 30 per cent in two (2) inches, in case 8-inch specimens cannot be secured.

Cold-bending Test.—One bar, one-half ($\frac{1}{2}$) inch thick, cut from each lot of Class A No. 1 bracing shall stand cold-bending double to an inner diameter of one (1) inch, the ends of the piece being brought parallel, without showing signs of fracture on the outside of the bent portion.

Cold-bending Test.—One bar, one-half ($\frac{1}{2}$) inch thick, cut from each lot of Class A No. 2 bracing shall stand cold-bending double to an inner diameter of one-half ($\frac{1}{2}$) inch, the ends of the piece being brought parallel, without showing signs of fracture on the outside of the bent portion.

Cold-bending Test.—One bar, one-half ($\frac{1}{2}$) inch thick, cut from each lot of Class A No. 3 bracing shall stand cold-bending flat on itself, the ends of the piece being brought parallel, without showing signs of fracture on the outside of the bent portion.

Opening and Closing Tests.—Angles, T bars, and other shapes are to be subjected to the following additional tests: A piece cut from one bar in twenty shall be opened out flat while cold, without showing cracks or flaws; a piece cut from another bar in the same lot shall be closed down on itself until the two sides touch, without showing cracks or flaws.

Inspection for Surface and other Defects.—Stay-rods and bracing must

be true to form, free from seams, hard spots, brittleness, injurious sand or scale marks, and injurious defects generally.

Boiler-plate used as boiler bracing shall be inspected as plate under the specifications for boiler-plate of the class required by the Machinery Specifications.

SPECIFICATIONS FOR RIVET-RODS AND FINISHED RIVETS.

Kind of Material.—*High-grade and Class A material* shall be made by the open-hearth process, shall not show more than thirty-five one-thousandths of 1 per cent of phosphorus nor more than three one-hundredths of 1 per cent of sulphur, and shall be of the best composition in other respects.

Class B material may be made by the open-hearth or Bessemer process, as ordered. The material shall not show more than nine one-hundredths of 1 per cent of phosphorus, nor more than six one-hundredths of 1 per cent of sulphur, and may be used on work specified as Class B, and in other work, if permitted by the Machinery Specifications.

The drillings for chemical analysis shall be taken from the cold-bending test specimens.

TENSILE, COLD-BENDING, AND QUENCHING TESTS.

Test Specimens.—If the total weight of rods rolled from a heat amounts to more than six tons, then the Naval Inspector shall select, at random, six tensile-test pieces, three cold-bending-test pieces, and three quenching-test pieces. If the total weight of rods rolled from a heat amounts to less than six tons, then the Naval Inspector shall select, at random, one tensile, one cold-bending, and one quenching test piece for each ton or part of ton.

Tensile, cold-bending, and quenching test specimens shall be cut from rods, finished in the rolls, and only one specimen is to be cut from each rod selected for test.

Should the specimen cut be so large in cross-section that its ultimate strength exceeds the capacity of the testing-machine, then the specimen is to be machined to the largest cross-section which can be broken in the testing-machine. If only one of these rods selected for test fails, the Naval Inspector shall select another rod from the same lot and put it through the same test as the one that failed, and if it is found satisfactory he shall pass the lot. The tensile test of sizes of rounds under five-eighths ($\frac{5}{8}$) of an inch in diameter shall be made on rounds of five-eighths ($\frac{5}{8}$) of an inch in diameter.

High-grade rivet-rods, of nickel steel, shall have at least 75,000 pounds

per square inch tensile strength, with an elastic limit of at least 40,000 pounds per square inch, and an elongation of at least 23 per cent in eight (8) inches.

Class A rivet-rods, of open-hearth steel, shall have at least 60,000 pounds per square inch tensile strength, with an elastic limit of at least 32,000 pounds per square inch, and an elongation of at least 26 per cent in eight (8) inches.

Class B rivet-rods, of open-hearth or Bessemer steel, shall have at least 54,000 pounds per square inch tensile strength, with an elastic limit of at least 30,000 pounds per square inch, and an elongation of at least 29 per cent in eight (8) inches.

Cold-bending Test.—*High-grade rods* shall stand cold-bending 180 degrees to an inner diameter equal to the thickness or diameter of the specimen.

Class A rods shall stand cold-bending 180 degrees to an inner diameter equal to one-half the thickness or diameter of the specimen.

Class B rods shall stand cold-bending 180 degrees, face to face.

All cold-bending tests will be satisfactory if the specimens show no cracks on the outside of the bend.

Quenching Test.—The specimen shall, after heating to a dark cherry-red, in daylight, be plunged into water of a temperature of 82° F., and shall then stand bending double to an inner diameter of one-half inch without showing cracks or flaws on the outside of the bent portion.

HAMMER TESTS ON FINISHED RIVETS.

Hammer Tests.—For each ton or less of finished rivets, made from the same heat, four rivets shall be selected, at random, by the Naval Inspector and submitted to the following tests:

(a) Two of these specimens to be flattened out cold to a thickness of one-half the original diameter or thickness of the part flattened without showing cracks.

(b) Two of these specimens to be flattened out hot to a thickness of one-third of the original diameter or thickness of the part flattened without showing cracks, the heat to be a cherry-red in daylight.

SURFACE INSPECTION.

Rivets shall be true to form, concentric, and free from injurious scale, fins, seams, and all other injurious defects.

Some very interesting experiments were made in 1898 by Maunsel White at the Bethlehem Iron Co. on the use of nickel-steel

for rivets.* The results showed that such steel could be subjected to the working heats without serious injury, and that a $\frac{3}{4}$ -inch nickel-steel rivet was about equal to a $1\frac{1}{8}$ -inch common steel rivet, thus saving a considerable portion of plate section. This material may replace ordinary steel for many uses.

Nickel-steel contains about 0.23% carbon, 0.02% sulphur, 0.55% manganese, 3.50% nickel, 0.02% phosphorus. The ultimate strength is about 84,000 pounds per square inch, the elastic limit about 52,000 to 55,000 pounds, the elongation about 20%, and reduction of area about 50% to 60%. The shearing strength is about equal to the tensile strength. The shearing strength of common steel rivets is about 43,000 to 46,000 pounds per square inch.

The tensile strength of mild steel, such as used for boiler-plates and furnaces, is appreciably affected by temperature. The results of experiments made by the United States Navy Department, 1888, as briefly stated by D. B. Morison (see *Cassier's Magazine*, August, 1897) were as below:

"The tensile strength of all steel varies between zero and 200 degrees Fahr.; the maximum strength is between 400 degrees and 600 degrees, and beyond 600 degrees the tensile strength decreases rapidly. Although the ultimate tensile strength increases from 200 to 600 degrees, the elastic limit steadily decreases from zero upwards, and steel, having an elastic limit of 35,000 pounds per square inch at zero, has its elastic limit reduced to 20,000 pounds per square inch at 600 degrees Fahr."

"Another point is that the higher carbon steels reach a temperature of maximum strength more abruptly and retain their highest strength over a less range of temperature than steels having a low percentage of carbon."

"In furnaces from $\frac{1}{2}$ inch to $\frac{3}{4}$ inch thick, under 200 pounds steam-pressure, the temperature of the plate, when clean, approaches a temperature of maximum tensile strength; therefore the use of hard, brittle steel is rendered dangerous, especially with furnaces of very rigid design."

Cast Steel. The quality and reliability of cast steel has been so advanced in the past few years as to render the material very

* See *Journal Am. Soc. Naval Engineers*, November, 1898.

popular for use in odd shapes, which were formerly made of wrought-iron forgings. The danger of flaws is fast disappearing, and so great a confidence is being placed in cast steel that it is superseding cast iron for many uses. As the demand increases, no doubt its manufacture will be vastly improved.

It is used for many fittings for water-tube boilers, manhole and handhole covers, ends for small steam- and mud-drums, grate-bars, etc.

Castings always should be designed with as even a thickness as possible throughout, and all sharp angles be rigidly avoided by using large fillets. If care in this particular be not taken, the casting is apt to be weak or defective from internal shrinkage stresses, and liable to crack.

As defects, blow-holes, etc., in steel castings are usually near the surface uppermost in the cast or at the centre of the upper end, care must be taken in making the mould, so that the defects can be most readily seen or be machined out while the piece is being shop-finished.

The tensile strength of cast steel is very high, but rapidly loses in elasticity. It is most difficult to keep the strength low and thus obtain a high elasticity. Cast steel with a tensile strength of 58,000 to 60,000 pounds should have an elongation of from 10 to 15 per cent in 8-inch test pieces. When the strength is between 60,000 and 70,000 pounds the elongation should never be less than 10 per cent, and may be as high as 18 per cent. For all important pieces the casting should be annealed to reduce the tensile strength and increase the elongation.

Cast-steel specimens of round or square section, $1\frac{1}{4}$ inches thick and less, should be capable of bending cold without fracture through an angle of 90 degrees, over an inner radius not exceeding $1\frac{1}{4}$ inches, when tensile strength is 60,000 pounds; and through an angle 10 degrees less for each increase of 5000 pounds in tensile strength.

Defects should not be made good by patching or by electric welding, unless done under careful supervision.

The requirements of the U. S. Navy Department, 1899, are in abstract as below:

Sound test pieces shall be taken in sufficient number to exhibit the character of the metal in the entire piece.

A lot shall consist of all castings from the heat annealed in the same

furnace charge. From each lot two tensile and one bending specimen shall be taken.

If there are any unsound test specimens coming from a casting, the inspector shall examine carefully to detect porosity or other unsoundness in the casting itself.

Steel for castings shall be made by either open-hearth or crucible process, and no casting shall show more than six one-hundredths (0.06) of one per cent of phosphorus and shall be of the best composition in other respects. All castings shall be annealed unless otherwise ordered.

The tensile strength of castings shall be at least 60,000 pounds per square inch, with an elongation of at least 15 per cent in eight inches for all castings for moving parts of machinery, and at least 10 per cent for all other castings. In thin castings, the inspector may require specimens two inches in length to show 20 per cent elongation with 62,000 pounds tensile for moving parts, and 15 per cent elongation with 60,000 pounds tensile for other parts.

A test to destruction may be substituted for the tensile test, in the case of small and unimportant castings. This test must show the material to be ductile and free from injurious defects and suitable for the purposes intended.

One bar or more from each important casting or one bar from each lot, one inch square, shall be bent cold without showing cracks or flaws, through an angle of 120 degrees for castings for moving parts of machinery, and 90 degrees for other castings, over a radius not greater than one and one-half inches. In cases where two-inch tensile specimens are used, a bending specimen one inch by one-half inch shall be bent cold through 150 degrees without showing cracks or flaws, for castings for moving parts of machinery, and 120 degrees for other castings.

All castings must be sound, free from brittleness, injurious roughness, sponginess, pitting, porosity, shrinkage and other cracks, cavities, foreign or other substances, and all other defects affecting their value. Particular search must be made at the points where the heads or risers join the castings, as unsoundness at this point is likely to extend into the castings.

Copper. The use of copper in boiler construction has gradually decreased, except for certain adjuncts, as steam-pipes, feed and blow-off pipes, gaskets, etc. It is still used to a limited extent for fire-box sheets in boilers of the locomotive type.

It is very ductile, and can be joined by brazing so as to be as strong as the original piece. The greatest care must be taken not to burn the copper while being prepared for brazing.

It does not corrode under the action of air or water, either fresh or salt, although certain waters containing free gases appear to act deleteriously upon it. It is extremely useful for forming alloys, as it is much improved by small quantities of other metals. It is expensive, has a low tensile strength and weakness to resist abrasion.

The more copper is worked the stronger it becomes within reasonable limits. Its strength depends upon its quality. Cast copper has a tensile strength of about 22,000 pounds, forged copper about 31,000 pounds and rolled copper about 33,000 pounds. Copper wire has a strength of about 60,000 pounds before annealing and 40,000 pounds after annealing.

For all calculations sheet copper can be figured at 30,000 pounds in default of full information. At high temperatures copper rapidly loses strength, becomes plastic at about 1330 degrees and melts at about 2000 degrees Fahrenheit.

Copper may be solid-drawn into pipes as large as 8 inches diameter.

From a paper by J. T. Milton read at the 14th Session of the Institution of Naval Architects, London, 1899, the following paragraphs are extracted:

"Hardening of copper may be produced in other ways than by direct tension. Copper wire is hardened by continual bending and straightening; sheet copper is hardened by hammering or by cold rolling; pipes may be hardened by planishing or by being hammered or bent whilst they are 'loaded,' and copper tubes are always hardened when they are drawn on a draw-bench either to a smaller diameter or a thinner gauge. In whatever way copper is hardened, its ductility is correspondingly lessened, and in all cases the hardening may be removed by 'annealing,' that is, by raising it to a bright-red heat, and either quenching it in water or allowing it to cool gradually.

"Commercial copper, as used for other than electrical purposes, is rarely pure, or even nearly pure. The effects of some of the common impurities, such as arsenic, nickel, and silver, are supposed not to be detrimental; while, on the other hand, antimony is objectionable, and bismuth, even in small traces, is exceedingly prejudicial. The usual workshop test for the quality of copper is to cut off a portion of the pipe or sheet and anneal it, when it should stand bending quite close without a sign of cracking. The edges also should stand thinning to a knife-edge without cracking when hammered to a scarf-joint form with a lap of about three or four times the thickness of the copper.

Brazing. Brazing-solder is composed of copper and zinc in about equal proportions; occasionally, however, one-half per cent of tin is added to the mixture. The mixed metal is first cast in iron ingot-moulds, then it is reheated to a certain temperature, considerably below red heat, at which it becomes brittle and is pounded up with an iron pestle and mortar. The addition of the small quantity of tin is said to facilitate the pounding. It thus appears that at a temperature intermediate between that of the steam and a red heat the solder becomes brittle and unfit to sustain any stress.

"It usually is considered that the brazing-solder, like copper, is not liable to corrosion, and in the majority of cases in which brazed copper steam-pipes have been cut up after many years of service the brazing is found to be in as good condition as the copper. In a few cases, however, the brazing of copper steam-pipes has been found to have deteriorated in use to an alarming extent. Attention was first drawn to this in the case of the fatal explosion of the steam-pipe of the S. S. "Prodano." After the official inquiry into the matter this case was investigated by Professor Arnold of Sheffield, whose report was published in *Engineering*, Vol. LXV, p. 468, and *The Engineer*, Vol. LXXXV, p. 363. Professor Arnold showed that the brazing in this and in another case submitted to him at the same time had deteriorated by the whole of the zinc in some parts of the solder becoming oxidized, the copper remaining in the form of a spongy metallic mass, the pores of which were filled with oxidized zinc. He attributed this result to electrolytic action set up by fatty acids produced in the boiler or in the steam-pipe from the decomposition of organic oils, as he found and separated these organic acids from the deteriorated solder. Since attention was drawn to these cases a few other steam-pipes have been found to have similarly depreciated in their brazing.

"It is worthy of note that experience with Muntz's metal exposed to the corrosive action of sea-water shows that a somewhat similar deterioration of the zinc takes place. It is said that this is prevented if a small quantity of tin is added to the mixture; but in the cases of the brazing-solder investigated by Professor Arnold, one specimen, which originally contained one-half per cent of tin, was equally affected to that composed of copper and zinc only."

Brass. Brass is an alloy of copper and zinc.

Bronze is an alloy of copper and tin, or of copper and tin with zinc or some other metal. Many of the bronzes are commonly called "brass."

In boiler construction brass is chiefly limited to tubes, flanges for pipes and boiler-mountings.

Brass boiler-tubes are used but little, except in some naval boilers and in feed-water heaters. It then, usually, has a composition of 68 per cent of best selected copper and 32 per cent of zinc. Its tensile strength exceeds 66,000 pounds per square inch.

Ordinary brass or yellow brass, composed of two parts of copper and one part of zinc, has a tensile strength of from 21,000 pounds to 27,000 pounds, and is consequently too soft for anything but ornamental purposes.

Boiler-mountings may be made of *Muntz's metal*, composed of three parts of copper and two of zinc, which is very ductile and has a strength of about 45,000 pounds; or of *naval brass*, which is made by adding about one per cent of tin to Muntz's metal. This naval brass is quite as strong as Muntz's metal and superior to it in ductility, and will resist the action of salt water.

The strengths of both are increased when rolled or forged and annealed, but are less than the above figure when cast.

CHAPTER VIII

BOILER DETAILS

The Shell. Strength of Shell, Longitudinally and Transversely. Factor of Safety. Rules for Thickness of Shell. Limits of Thickness. Arrangements of Plates. The Ends. Rules for Thickness of Heads. Flat Surfaces. Rules for Flat Surfaces. Flues. Strengthening Rings. Corrugated and Ribbed Flues. Rules for Flues and Liners. Tubes. Rules for Thickness. Stays. Rules for Stays. Girders. Combustion-chamber. Riveting. Welding. Setting. Bridge Wall. Split Bridge.

The Shell. The pressure of steam, or of any gas, enclosed in a cylindrical shell is exerted equally in all directions, and is at right angles to the surface and therefore along radial lines. The pressure tends to enlarge the vessel and to keep it in the true cylindrical form. It is resisted by the strength of the metal, the tendency being to create rupture longitudinally.

As the metal is never absolutely homogeneous, the rupture commences at some weak point, but does not always follow the line of least resistance, because the final failure takes place so suddenly that not only does time become an element, but also new and undeterminable stresses are introduced.

Let AB in Fig. 37 be any diameter of the circle whose centre is at O , and which may represent the section of a cylindrical boiler-shell. Let xy represent a very small length of the section, which may be considered as straight without any appreciable error. The pressure exerted on this area xy can be represented in direction and magnitude by pxy , p being the pressure per unit of area. Resolve this force into its two components, one perpendicular and one parallel to the diameter AB , and denote the angle between the direction of xy and AB by α . The vertical component is then $p \cdot xy \cos \alpha$. But $xy \cos \alpha$ is equal to the projection of xy on the diameter AB . If the same analysis were made for every point of the semi-circumference, then the algebraic sum would be $\sum p xy \cos \alpha$ equal to

$p \cdot AB$. In other words, the sum of all the perpendicular components is equal to the unit of pressure times the diameter, and this is the force tending to produce rupture along the longitudinal lines at A and B .

The parallel components, being equally distributed on each side, neutralize one another, are therefore inert and have no effect on the stresses at A and B .

The cylinder resists rupture by an amount equal to the area of the metal at A and B times the tensile strength of the

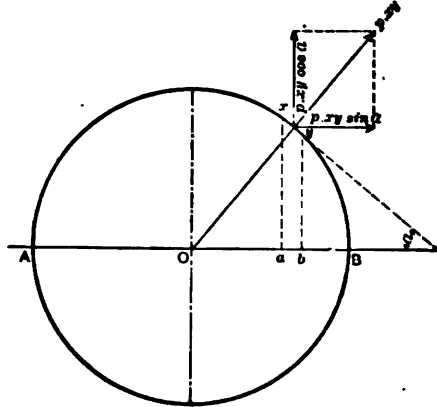


FIG. 37.—Strength of shell to resist bursting pressure.

material. (This statement is only true for cylinders whose thickness is very small in comparison with the diameter, and the following analysis fails for all cases where the diameter is small and the thickness large, as in cannon or hydraulic cylinders. This is due to the elasticity of the material, which in thick shells would allow the inner layers to be stretched to an injurious amount before the outer ones had reached their elastic limit.)

At the moment of rupture, denoting the bursting pressure by P pounds per unit of force, the force of the gas and the resisting strength of the shell are equal, and consequently

$$P \times D \times \text{Length} = 2 \times t \times c \times \text{Length}.$$

$$P = \frac{2tc}{D},$$

$$t = \frac{PD}{2c},$$

in which c denotes the tensile strength.

It is to be noted that the strength of a shell to resist bursting is independent of the length; but, as will be seen later, the length plays a very important part in the strength when contraction and

expansion are considered, as also when the pressure is external to the shell tending to produce failure by collapsing.

In a boiler-shell with lap-jointed plates the perfect cylindrical form cannot be obtained, but since the distortion is very small, the weakness is more due to the uneven distribution of the stress on the rivets of the joint than to the lack of a perfect cylindrical section.

The ends of the cylinder, especially when flat, act like stays, and must give it increased strength. How much increase in strength or how far from the ends does this end influence extend is unknown, but in boilers having a short longitudinal dimension compared to diameter the influence may be considerable. (Reference is made to a paper by J. G. Spence, Trans. North-east Coast Institution of Engineers and Shipbuilders, and discussion in Engineering, 1891 *et seq.*).

In oval boilers this additional strength due to the ends is important, but on account of the high modern pressures oval boilers are seldom used.

In practice no allowance is made for the strength imparted by the ends, due to its uncertain character. Since undeterminable conditions often exist, due to faults in staying, poor joints, flaws and corrosion, all of which cannot be allowed for with accuracy, this increase in strength is permitted to offset, in whole or in part, these sources of weakness.

The stress tending to cause rupture transversely or in a ring direction around the boiler is the pressure on the boiler-head. The resisting strength is evidently that of the circular section of the metal.

At the moment of rupture these forces are equal, and, using the same letters as before, can be expressed thus:

$$P \frac{\pi D^2}{4} = \pi (D+t) t c.$$

$$P \frac{D}{4} = \left(1 + \frac{t}{D}\right) t c.$$

Neglecting $\frac{t}{D}$, since it is always a very small fraction,

$$P = \frac{4tc}{D} \quad \text{and} \quad t = \frac{PD}{4c}.$$

On comparing this result with that obtained for longitudinal rupture, it will be noted that any shell of uniform thickness is twice as strong transversely as longitudinally, and that if the metal be calculated for thickness in the latter direction it will be amply heavy to resist in the former direction. On account of wear due to buckling from expansion and the consequent tendency to groove and corrode, boilers often fail on the ring seams, so that these seams demand as much care in proportioning as the longitudinal ones, although not subjected to so heavy a stress.

Due to the difference in stress, it is quite common to rivet the longitudinal seams more strongly than the ring seams.

External Pressure on a Stayed Shell. In cases where the pressure is external to a cylinder which is supported by stay-bolts, as for example the inner shell or furnace of a vertical boiler, it would be well to call attention to the conclusions stated in *Locomotive*, March, 1892:

(1) The stresses in the plates of a curved water-leg are never greatly different (at the usual working pressures) from those that prevail in a similarly designed flat-stayed surface; (2) The curved form of the leg does, however, cause the tension on the outer sheet to be somewhat greater than it would be in a flat leg; (3) The stress on the inner sheet is never a compression, but always a tension; (4) This tension on the inner sheet will differ (usually by a small amount) from the tension on a similar flat stay-bolted sheet, being sometimes greater and sometimes less, according to the designs and proportions of the water-leg; and (5) The curvature of the leg causes the stress on the stay-bolts to be somewhat less than it would be on a similar flat leg.

Factor of Safety. The proper factor to use in determining the thickness of a boiler-shell is open to dispute. It is evident that it should be as small as safety will permit, and its selection should depend on the quality of metal used, on the methods adopted in construction and on the efficiency of the workmanship employed. Probably under good inspection and with careful builders a factor of safety of three is sufficient, but this factor should be taken on the weakest part, which is usually the joint, and not on the strength of the shell plating.

Rules for Thickness of Shell. There can be little question as to the advisability of proportioning the thickness on the elastic limit of the metal rather than on the ultimate strength.

While a boiler may not burst when the elastic limit has been exceeded, still the shell has practically failed, due to the permanent distortion which has taken place and which will reduce the actual strength. There is, however, a commercial disadvantage in the use of elastic limit, since so many boilers are made of sheets of which this property is not known, although the plates could be as easily tested for the elastic limit as for ultimate tensile strength.

1. *Rule of U. S. Board of Supervising Inspectors of Steam-vessels.*—Multiply one-sixth ($\frac{1}{6}$) of the lowest tensile strength found stamped on any plate in the cylindrical shell by the thickness—expressed in inches or parts of an inch—of the thinnest plate in the same cylindrical shell, and divide by the radius or half diameter—also expressed in inches—and the result will be the pressure allowable per square inch of surface for single-riveting, to which add 20 per cent for double-riveting, when all the rivet holes in the shell of such boiler have been “fairly drilled” and no part of such hole has been punched.

This rule is the one most commonly employed in this country, but is very defective. It does not consider workmanship or the strength of joint, which is the real strength of the shell. The same thickness would be used for a well-proportioned joint as for a poor one.

2. *Rule of Lloyd's Registry (British).*—For steel cylindrical boiler-shells,

$$\left. \begin{array}{l} \text{Working pressure in pounds} \\ \text{per square inch} \end{array} \right\} = p = \frac{\text{constant} \times (t-2) \times B}{D},$$

in which D denotes mean diameter of shell in inches;

t “ thickness of plate in sixteenths of an inch
expressed as a whole number;

Constant is 21 when the longitudinal seams are fitted
with double butt-straps of equal width;

Constant is 20.25 when they are fitted with double butt-
straps of unequal width, only covering
on one side the reduced section of
plate at the outer lines of rivets;

Constant is 19.5 when the longitudinal seams are lap-joints;
 B denotes the least percentage of strength of longitudinal
joint.

Note.—For shell plates of super-heaters or steam-chests enclosed in the uptake or exposed to the direction of the flame the constants should be two-thirds of those given above. The material to have an ultimate tensile strength of not less than 58,240 pounds and not more than 67,200 pounds per square inch of section.

The value of B can be found as follows, the least value being taken:

$$\text{For percentage of plate at joint, } B = \frac{p-d}{p} \times 100,$$

$$\text{“ “ “ rivets “ “ } B = \frac{n \times a}{p \times t} \times F,$$

in which t denotes thickness of plate in inches;

p	“	pitch of rivets in inches;
d	“	diameter of rivet holes in inches;
a	“	sectional area of rivets in inches;
n	“	number of rows of rivets;
F	“	100 when rivets are iron and plates are iron, holes punched;
F	“	90 when rivets are iron and plates are iron, holes drilled;
F	“	85 when rivets are steel and plates are steel, holes drilled;
F	“	70 when rivets are iron and plates are steel, holes drilled.

Note.—If rivets are in double shear, use $1.75a$ instead of a .

There are other rules, such as Board of Trade (British), Bureau Veritas, German Lloyds, etc. Certain cities have rules for local inspection, as have the various boiler-insurance companies. Many authors have advanced rules of their own, but they are chiefly modifications of the principal ones above mentioned.

Reference is made to a criticism of the rules used for marine boilers (which applies equally to land boilers) in a paper by Nelson Foley, published in Transactions of Division of Marine and Naval Engineering, Chicago Engineering Congress, 1893, and to a paper by the author, Trans. Am. Soc. Mechanical Engineers, Vol. XXII, p. 127, 1901.

Many engineers place the minimum thickness of shell for externally fired boilers at $\frac{3}{8}$ of an inch, because thin sheets exposed to sudden extreme variations of temperature lose in elasticity, and will corrode as much as thick ones, so that the percentage of loss of strength will be much greater.

The maximum thickness is usually limited to $\frac{5}{8}$ or $\frac{3}{4}$ inch. The thickness is controlled by the liability to burn in externally fired boilers. In shells not exposed to the fire the thickness is only limited by the appliances of the boiler-shop. There are few boiler-shops equipped to properly handle sheets exceeding $1\frac{1}{2}$ inches in thickness. Also, thick sheets are difficult to rivet, as the size of rivet is limited, and small rivets in thick sheets necessitate too much cutting of the plates in order to procure a high efficiency of joint.

When the shell of externally fired boilers is very thick, the life of the boiler may be prolonged by constructing a fire-brick arch over the furnace. Such an arch assists the combustion by preventing the products from becoming chilled, and also shields the crown-sheet from impingement of cold air through the fire-doors. When used, however, care must be taken not to bury the boiler in masonry, but arrange for inspection of the covered portion of the boiler. Oftentimes this only can be done by so constructing the arch that it will not come into actual contact with the shell, and that it may be removed for inspection and repair without disarranging the rest of the setting.

Arrange the sheets so as to keep the seams as far as possible from the direct action of the fire. This cannot always be accomplished with ring-seams of externally fired boilers, except by the use of very large sheets, the cost of which may be prohibitive. In such boilers the longitudinal seams can usually be placed out of reach of the fire without complication. When lap-seams must be exposed to the passage of the hot gases, arrange so that the gases do not strike against the edge of the lapping plate. In like manner arrange the lap-seams so that the circulation or convection currents will not strike against a lapping edge, upon which the sediment may lodge. The lack of this precaution has often caused a crown-plate to bulge downward.

Place the plates so that the direction in which they were rolled shall be circumferential, that the direction of greatest strength shall be in line with the greatest stress.

Make use of as large sheets as possible, so as to reduce the number of seams. Boiler-plates seldom are made larger than 121 inches in width, but can be procured of any reasonable length. The length usually is limited to about 33 feet, which is the length of the platform of a freight-car. Plates are not available for their full length or width, as the edges of the sheets are usually the weakest parts and are most liable to carry defects, and it is near the edges that the line of rivets must come. Plates should be trimmed off on each side by at least $3\frac{1}{2}$ to 4 inches for careful work.

Arrange the longitudinal seams so as to break continuity at the circumferential seams, as thereby increased shell strength is obtained. These longitudinal seams in adjacent courses of plates should be separated as far apart as can be arranged conveniently, but when prevented by other reasons the centre lines of the seams may be as near as the pitch of three rivets. At such places three plates must overlap, and the corner of the middle plate is forged thin, tapering like a wedge, so that the other plates may come gradually into contact and be calked.

The successive courses in long boilers are best made parallel; that is, one course is an outer course, the second laps under the first and third, forming an inner course, and so on alternately inside and outside. By this arrangement the brick setting is also least apt to be disturbed by the expansion of the boiler. Sometimes the courses are parallel, but each successive course is arranged as an inner course to the one preceding and as an outer course to the one following. This method is adopted in some locomotive boilers, the largest course being at the fire-box end. Occasionally the courses are arranged conically; that is, as an outside course to the one preceding and as an inner course to the one following. The objection to this latter method is the difficulty of cutting the sheets, as they will not be rectangular, and the danger of the rivet-holes not matching fair. They are also more liable to disarrangement from expansion and contraction than when the sheets are parallel.

In vertical boilers it is best to arrange the lap of horizontal ring-seams to face downward on the inside, so as to prevent sediment from catching.

Whatever arrangement is adopted, the shell should be so placed as to drain itself when required. This is accomplished by setting the shell lower at the end where the bottom blow-off is located. In

general a difference of from 1 inch to 2 inches in 25 feet of length is sufficient.

The Heads. It is obvious that the best form for resisting internal pressure is the sphere; and, further, the pressure will always tend to preserve the true spherical form, which will therefore be self-supporting.

The pressure tending to burst a sphere along the intersection of any plane passing through its centre is measured by the pressure per unit of surface times the area of such a plane, which force may be expressed thus:

$$P \frac{\pi D^2}{4}.$$

The resistance to rupture is the strength of the metal at the place of intersection of the plane, which may be expressed thus:

$$\pi(D+t)tc.$$

At the instant of rupture these quantities are equal.

$$P \frac{\pi D^2}{4} = \pi(D+t)tc.$$

Dividing by πD , and neglecting $\frac{t}{D}$, which is always a small fraction in boiler construction, the expressions for bursting pressure and thickness become

$$P = \frac{4tc}{D} \quad \text{and} \quad t = \frac{PD}{4c}.$$

These are the same results as were obtained for the transverse strength of the cylinder; so that the sphere is twice as strong as is the cylinder longitudinally, when the diameters are equal.

Advantage of this fact, as well as of the self-sustaining property, is taken in making the heads of many boilers portions of a sphere. When the diameter of the sphere of which the head is part is twice the diameter of the cylinder, the strength of shell and head will be equal to resist bursting. On the other hand, the less the diameter of the spherical heads, the less will be their efficiency in strengthening the cylindrical shell.

In all plain cylindrical boilers the heads are always thus made.

and such heads are said to be "cambered," or "dished," or "bumped," see Fig. 9.

All boiler-heads would be made cambered if it were not for the difficulty of obtaining tight joints with tubes and flues. In many Scotch or drum boilers the heads are made part cylindrical above the top row of tubes, thus avoiding heavy stays in the steam-space, see Fig. 22.

The heads, whether flat or cambered, are best fastened to the shell by flanging the end sheets so as to pass on the inside of the shell plating. The flanging should be over a radius about $2\frac{1}{2}$ times the thickness. The flange should be single-riveted to the shell when the circumferential seams are double-riveted, and should be double-riveted when those seams are treble-riveted or quadruple-riveted; although every case should be considered by itself, as much depends on the bracing of the head by the stays.

When flat heads are used, one end is best flanged in and the other out, so that both may be closed by the power-riveter. When both heads are flanged inward, the head put on last must be hand-riveted. With externally fired, return-tubular or flue boilers the rear heads should be flanged in, so as not to expose the unprotected thin edge of the joint to the action of the hot gases. With an extended smoke-box the front head may be flanged out. The head may be fastened by an angle, but flanging is much better, being less liable to suffer from corrosion or grooving.

When an angle is used, it should be put on the outside of the shell, at least at one end, and better on both unless the back head be exposed to the returning hot gases, as in Lancashire and Cornish boilers. The internal angle forms too stiff an attachment, since the distance from the row of rivets to the nearest flue is so short. The external angle makes this distance longer, and allows the head to spring so as to relieve the joint and prevent grooving, when the pressure is brought upon the head by expansion of the flue. It is very desirable that flat heads be not made too stiff, in order that they may spring sufficiently to relieve the local strains caused by expansion and contraction.

It is best to make the head out of one sheet when possible. In large boilers this is often impossible, and the head must be made of two or more pieces joined together by rivets. The seam is usually made lapped, and its position selected to suit the gen-

eral design. No rule can be stated for single- or double-riveting this joint, selection depending on the judgment of the designer and the method of staying. Often this lap can be so arranged as to receive the ends of some of the stays, and thus dispense with washers required by the rules for flat surfaces.

Rules for Thickness of Heads. For "cambered" or "dished" heads the thickness or pressure may be calculated by the rules given for cylinders, but using the diameter of the sphere, of which the head forms a part, as the value of D .

The U. S. Board of Supervising Inspectors of Steam-vessels states that the pressure of steam per square inch allowed shall be found by multiplying the thickness of plate by one-sixth of the tensile strength, and dividing by one-half of the radius to which the head is bumped. Where the heads are concaved, multiply the pressure per square inch found for similar bumped heads by six-tenths, and the result will be the steam-pressure allowable.

On unstayed flat heads, when flanged and made of steel, of wrought-iron, or of cast-steel, the pressure per square inch allowable is determined by multiplying the thickness of plate in inches by one-sixth the tensile strength in pounds, and dividing by the area of the head in square inches multiplied by nine one-hundredths.

Flat heads supported by stays are proportioned by the rules for flat surfaces.

Flat Surfaces. Many parts of a boiler must necessarily be made of flat plates. In order to make them self-supporting, they need be so thick as to be impracticable. In consequence they are strengthened or supported in various ways, but most commonly by stays.

The formula adopted is based upon that for beams; the portion of the plate lying between the rows of stays being taken as equivalent to a beam, rectangular in section, fixed at the ends and uniformly loaded, any extra strength due to end connections being neglected.

Flat surfaces should be avoided as much as possible, as not only being an element weak *per se*, but also rendering difficult the cleaning and inspection from the numerous stays that are required.

The end plates of boilers, in the steam-space, where not subjected to contact with hot gases, are frequently strengthened by riveting to them a "doubling-plate." Where the rivets are not spaced

farther apart than the distance allowed for bolts on a flat surface, the plate is considered of extra thickness, so that the stays may be spaced far enough apart to permit a man to pass between them. (See rules given below.) Such doubling-plates are used chiefly on Scotch boilers designed for high pressures. In ordinary land or stationary boilers the heads are stayed to the shell by gusset-plates and diagonal stays, which do not interfere with entrance through the manhole above the tubes, as would the "through and through" stays from head to head.

The flat tube-sheets are sufficiently supported by the tubes when they are expanded and have ends "beaded over" or "flared." When very high pressures are used it is well to make some of the tubes as stay-tubes and thus provide additional support. This is the practice in many marine boilers. The space between the nests of tubes, and between outer tubes and side of shell, as also between furnace-flues, must be stayed, or thickness made sufficient for the pressure required. These latter spaces often are strengthened by angles or tees riveted to the flat surfaces. (See rules below and also "Stays.") Tube-sheets should never be less than $\frac{3}{8}$ -inch thick, so that the tubes may be cut out and new ones put in without injury to the plate.

Rules for Flat Surfaces.

1. *U. S. Board of Supervising Inspectors of Steam-vessels.*

The flat surface at back connection or back end of boilers may be stayed by the use of a tube, the ends of which being expanded in holes in each sheet beaded and further secured by a bolt passing through the tube and secured by a nut. An allowance of steam shall be given from the outside diameter of pipe. For instance, if the pipe used be $1\frac{1}{2}$ inches diameter outside, with a $1\frac{1}{4}$ -inch bolt through it, the allowance will be the same as if a $1\frac{1}{2}$ -inch bolt were used in lieu of the pipe and bolt. And no brace or stay-bolt used in a marine boiler will be allowed to be placed more than $10\frac{1}{2}$ inches from centre to centre to brace flat surfaces on fire-boxes, furnaces, and back connections; nor on these at a greater distance than will be determined by the following formulas.

Flat surfaces on heads of boilers may be stiffened with doubling-plate, tees, or angles.

The working pressure allowed on flat surfaces fitted with screw stay-bolts riveted over, screw stay-bolts and nuts, or plain bolt with

single nut and socket, or riveted head and socket, will be determined by the following rule:

When plates $\frac{1}{4}$ inch thick and under are used in the construction of marine boilers, using 112 as a constant, multiply this by the square of the thickness of plate in sixteenths of an inch. Divide this product by the square of the pitch or distance from centre to centre of stay-bolt.

EXAMPLE. Plate $\frac{1}{4}$ inch thick with socket bolts or stay, 6-inch centre, would be 112, the constant, multiplied by the square of 7, the thickness of the plates in sixteenths, which is 49, would give 5488, which, divided by the square of 6, which is 36, being the distance from centre to centre of stays or the pitch, would be 152, the working pressure allowed, provided the strain on stay or bolt does not exceed 6000 pounds per square inch of section.

Plates $\frac{1}{4}$ inch thick, stay-bolts spaced 4-inch centre = $\frac{112 \times 16}{16} = 112$ pounds W. P.

Plates $\frac{1}{4}$ inch thick, stay-bolts spaced 5-inch centre = $\frac{112 \times 25}{25} = 112$ pounds W. P.

Plates $\frac{1}{4}$ inch thick, stay-bolts spaced 6-inch centre = $\frac{112 \times 25}{36} = 77$ pounds W. P.

Plates $\frac{1}{2}$ inch thick, stay-bolts spaced 6-inch centre = $\frac{112 \times 36}{36} = 112$ pounds W. P.

Plates above $\frac{1}{4}$ inch thick the pressure will be determined by the same rule, excepting the constant will be 120; then a plate $\frac{1}{2}$ inch thick, stays spaced 7 inches from centre, would be as follows: 120, the constant, multiplied by 64, the square of thickness in sixteenths of an inch, equals 7680, which, divided by the square of 7 inches (distance from centre to centre of stays), which is 49, would give 156 pounds W. P.

Plates $\frac{3}{4}$ or $1\frac{1}{4}$ of an inch thick, spaced $10\frac{1}{2}$ inches, would be

$$\frac{120 \times 144}{110.25} = 156 \text{ pounds W. P.}$$

On other flat surfaces there may be used stay-bolts with ends threaded, having nuts on same, both on the outside and inside of plates. The working pressure allowed would be as follows:

A constant 140, multiplied by the square of the thickness of plate in sixteenths of an inch, this product divided by the pitch or distance of bolts from centre to centre, squared, gives working pressure.

EXAMPLE. A plate $\frac{3}{4}$ inch thick, supported by bolts 14 inches, would be

$$\frac{140 \times 144}{196} = 102 \text{ pounds W. P.}$$

Same thickness of plate, with bolts 12-inch centres, would be

$$\frac{140 \times 144}{144} = 140 \text{ pounds W. P.}$$

Flat part of boiler-head plates when braced with bolts having double nuts and a washer at least one-half the thickness of head, where washers are riveted to the outside of the head, and of a size equal to $\frac{1}{4}$ of the pitch of stay-bolts, or where heads have a stiffening-plate, either on inside or outside, covering the area braced, will equal the thickness of head and washers; the head and stiffening-plate being riveted together with rivets spaced and of sufficient sectional area of rivets as determined by this section for socket-bolts, shall be allowed a constant of 200, rivets to be spaced by thickness of washer on the stiffening-plate. Boiler-heads so reinforced will be allowed a thickness to compute pressure allowed of 80 per cent of the combined thickness of head and washer or head and stiffening-plate.

EXAMPLE. A boiler-head plate $\frac{3}{4}$ inch thick, with washers $\frac{3}{8}$ inch thick, and $12\frac{1}{2}$ inches square, supported by bolts 14-inch centres, would be allowed a working steam-pressure as follows:

Thickness of plate and washer equals $\frac{3}{4} + \frac{3}{8} = \frac{9}{8}$ inches; 80 per cent of which combined thickness equals $\frac{9}{8} \times \frac{80}{100}$ inch = .9 inch = 14.4 sixteenths of an inch.

Then, by rule,

$$\frac{200 \times 14.4^2}{14^2} = \frac{200 \times 207.36}{196} = 211 \text{ pounds W. P.}$$

Plates fitted with double angle-iron and riveted to plate with leaf at least two-thirds thickness of plate and depth at least one-fourth of the pitch would be allowed the same pressure as determined by formula for plate with washer riveted on.

EXAMPLE. A boiler-head plate $\frac{3}{4}$ inch thick, supported by angle-iron $\frac{1}{2}$ inch thick and 3.5 inches depth of leaf, and with bolts 14-inch centres, would be allowed a working steam pressure as follows:

Thickness of head and leaf of angle-iron equals $\frac{3}{4} + \frac{1}{2} = \frac{5}{4}$ inches; 80 per cent of which combined thickness equals $\frac{5}{4} \times \frac{80}{100}$ inches = 1 inch = 16 sixteenths inch.

Then, by rule,

$$\frac{200 \times 16^2}{14^2} = \frac{200 \times 256}{196} = 261 \text{ pounds W. P.}$$

But no flat surface shall be unsupported at a greater distance in any case than 18 inches, and such flat surfaces shall not be of less strength than the shell of the boiler, and able to resist the same strain and pressure to the square inch. In allowing the strain on a screw stay-bolt, the diameter of the same shall be determined by the diameter at the bottom of the thread.

2. Lloyd's Registry (British).

$$\text{Working pressure, pounds per square inch} = \frac{C \times t^2}{S^2},$$

in which t denotes thickness of plate in sixteenths of an inch;

- S " greatest pitch in inches, between centre of stays;
- C " 90 for plates $\frac{1}{8}$ thick and less, fitted with screw-stays with riveted heads;
- C " 100 for plates over $\frac{1}{8}$, fitted with screw-stays with riveted heads;
- C " 110 for plates $\frac{1}{8}$ and less, fitted with screw-stays and nuts;
- C " 120 for iron plates over $\frac{1}{8}$, and for steel plates over $\frac{1}{8}$ and under $\frac{1}{4}$, fitted with screw-stays and nuts;
- C " 135 for steel plates $\frac{1}{4}$ and above, fitted with screw-stays and nuts;
- C " 140 for iron plates fitted with stays with double nuts;
- C " 150 for iron plates fitted with stays with double nuts, and washers outside the plates, of at least $\frac{1}{2}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates;
- C " 160 for iron plates fitted with stays with double nuts, and washers riveted to the outside of the plates, of at least $\frac{2}{3}$ of the pitch in diameter and $\frac{1}{2}$ the thickness of the plates;
- C " 175 for iron plates fitted with stays with double nuts, and washers riveted to the outside of the plates, when the washers are at least $\frac{2}{3}$ of the pitch in diameter and of the same thickness as the plates.

For iron plates fitted with stays with double nuts, and doubling-strips riveted to the outside of the plates, of the same thickness as the plates, and of a width equal to $\frac{1}{2}$ the distance between the rows of stays,

C may be taken as 175 if S is taken to be the distance between the rows, and 190 when S is taken to be the pitch between the stays in the rows.

For steel plates other than those for combustion chambers the values of C may be increased as follows:

$C =$	140	increased to	175
	150	"	185
	160	"	200
	175	"	220
	190	"	240

If flat plates are strengthened with doubling plates securely riveted to them, having a thickness of not less than $\frac{2}{3}$ of that of the plates, the strength to be taken from

$$\text{Working pressure in pounds per square inch} = \frac{C \times \left(t + \frac{t'}{2}\right)^2}{S^2},$$

in which t' denotes the thickness of doubling-plates in sixteenths of an inch, expressed as a whole number.

Note. In the case of front plates of boilers in the steam space the values of C should be reduced 20 per cent, unless the plates are guarded from the direct action of the heat.

For steel tube-plates in the nest of tubes the strength to be taken from

$$\text{Working pressure in pounds per square inch} = \frac{140 \times t^2}{S^2},$$

in which S denotes mean pitch of stay-tubes from centre to centre.

For the wide water-spaces between nests of tubes the strength to be taken from

$$\text{Working pressure in pounds per square inch} = \frac{C \times t^2}{S^2},$$

in which S denotes the horizontal distance from centre to centre of the bounding rows of tubes;

- | | | | |
|-----|---|-----|---|
| C | " | 120 | when the stay-tubes are pitched with two plain tubes between them, and are not fitted with nuts outside the plates; |
| C | " | 130 | if they are fitted with nuts outside the plates; |
| C | " | 140 | if each alternate tube is a stay-tube not fitted with nuts; |
| C | " | 150 | if they are fitted with nuts outside the plates; |

in which C denotes 160 if every tube in these rows is a stay-tube and not fitted with nuts;

C " 170 if every tube in these rows is a stay-tube and each alternate stay-tube is fitted with nuts outside the plates.

The thickness of tube plates of combustion-chambers in cases where the pressure on the top of the chambers is borne by these plates is not to be less than that given by the following rule:

$$t = \frac{p \times w \times s}{1750 \times (s - d)},$$

in which t denotes thickness of tube-plate in sixteenths of an inch;

p " working pressure in pounds per square inch;
 w " width of combustion-chamber over plates in inches;
 s " horizontal pitch of tubes in inches;
 d " inside diameter of plain tubes in inches.

Flues. In internally fired boilers the grates are often constructed inside of large flues; and in many others flues are provided to carry the products of combustion and add heating surface.

On all such flues the pressure is external, tending to cause failure by collapsing. The flue may be considered as an arch, and since the pressure is the same on all points, the section should be circular in order to present a uniform resistance.

Should the section not be perfectly cylindrical, the external pressure will tend to increase the flatness and finally cause collapse. Thus external pressure acts in an opposite way from internal pressure, since it tends to aggravate rather than diminish the weakness due to any departure from the true cylindrical form. Therefore with flues the strength imparted by the end attachment becomes an important element to assist in keeping the form; and the greater the distance from the end, the less will this extra strength become. From the foregoing remarks it is evident that length must be an element in the strength of any flue subjected to external pressure.

Owing to the complexity of the problem, it is practically impossible to calculate the stresses existing in any flue, and the formulæ accepted by engineers are based on actual experiment.

The formula of Sir William Fairbairn is:

$$P = \frac{806,300 \times t^{2.19}}{Ld},$$

in which P denotes the collapsing pressure per square inch;

t " " thickness in inches;

d " " diameter " " ;

L " " length in feet;

The strength probably does not vary inversely as the length, but this rule is found to give sufficiently accurate results, provided the circular form is not departed from by more than twice the thickness of the plates, and the flue is not abnormally long.

For the sake of simplicity it is customary to use t^2 , rather than $t^{2.19}$. As in all boiler work t is a fraction, the use of t^2 will give a slightly higher collapsing pressure, which is provided against by the use of the larger factor of safety always adopted for flues.

The heat is always greater on the top than on the bottom of any furnace-flue, and the effect of expansion is to make the flue ellipsoidal or egg shaped in section. The external pressure on the flattened sides tends to increase this distortion, and the resistance is found to be inversely as the largest radius of curvature. Similarly, in flues that are made elliptical in section the above formula may be used by substituting for d the diameter of the larger circle of curvature, or by making d equal $\frac{2x^2}{y}$, in which x and y are respectively the major and minor axes of the ellipse.

Owing to the high pressures now so general, elliptical flues cannot be recommended, and ought not to be used except in special cases.

The simplest flues are made of sheets rolled into proper form, and the edges lap welded, lapped and riveted, or butted and joined by straps. The latter method is not popular, due to the large area of double thickness of metal. Riveted flues are cheaper than welded ones. When flues are less than seven inches in inside diameter, they are very difficult to rivet to the flange on the end plate, unless the rivets can be driven from the outside of flue, which is seldom the case on account of the proximity of the shell or other flues.

When long, flues must be built up with successive courses of plates. The ring seams when lapped increase the strength, and, within moderate limits, the length in the formula may be taken as the length between such ring seams. The successive courses should never be arranged with the longitudinal seams in line, but such seams should be made to break continuity as far as possible. However, the pitch of four rivet spaces will suffice.

In flues of large diameter and great length it would be necessary to use very thick metal. In order to keep the metal thin, the flue must be strengthened at intervals, so that the length may be taken as the length of such interval.

There are various ways of stiffening plain flues.

1. *By Lapping the Various Courses of Plating.* This method is little used, due to the uncertain amount of additional strength attributable to the laps. Except with large factors of safety, such flues are usually treated as plain flues.

2. *By Angle or Tee Rings.* Such rings are placed around the flue on the water side, and should be spaced off from the flue by at least one inch, so as to prevent a double thickness of metal (Figs. 38 and 39). The stays or bolts are passed through ferrules or distance pieces, so as to keep the ring firmly in position. Angles are

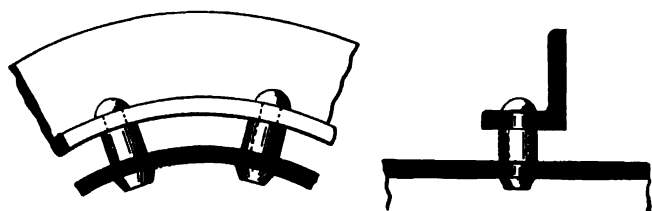


FIG. 38.—Flue Strengthening by Angle Ring.

generally preferred to tees; and angles 3 by 2½ by ¾ inches are heavy enough except for extremely high pressures. Plain rings may be used, riveted together, with stays passing between them.

These methods are not desirable for furnace-flues, due to the tendency of the flue to expand more than the ring, and the danger of breaking off the heads of the stays. Reference is made to failure of the furnaces of the "Bergen," Transactions Am. Soc. M. E., Vol. XI, 1890, p. 423.

There are no fixed rules for spacing the stays, but care must be taken to keep them close enough to lend sufficient support. These stay-rivets should have a diameter at least equal to one and one-half times the thickness of flue, and be spaced not over six inches apart, centre to centre.

When flues are close together these rings may be placed at varying distances on adjacent flues so as not to come opposite.

Care must also be taken not to allow them to interfere with convection currents, or to permit them to collect sediment or scale.

3. *By Tee Ring Joint.* Such rings may be used by riveting to each leaf the butting ends of the flue sheets (Fig. 40). The butts

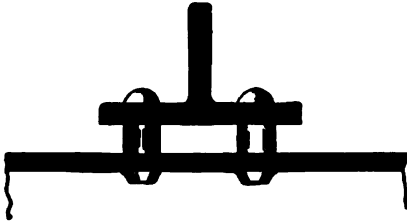


FIG. 39.—Flue Strengthening by Tee Ring.

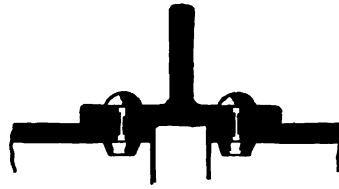


FIG. 40.—Flue Strengthening by Tee Ring Joint.

should be spaced apart far enough to calk the edge. This space must vary with the thickness, but for ordinary plates about one inch will suffice. The size of tee must be such that its thickness equals that of plates, and the web be three inches for ordinary pressures.

The objection is the double thickness exposed to the hot gases.

4. *By Flanging the Edges.* The edges may be flanged up so as to lap the rivet, or one edge be flanged up so as to lap a course of larger diameter (Figs. 41 and 42). Single-riveting is sufficient.



FIG. 41.—Flue Strengthening by Flanging.



FIG. 42.—Flue Strengthening by Flanging.

These methods are seldom used. The former is very expensive, as it necessitates a flange on both sheets.

5. *By the Bowling Hoop.* This method has met with much success (Fig. 43). It allows a certain amount of longitudinal elasticity, but is expensive and has two joints in the fire, together with a double thickness of metal. It is used less than formerly, having given way to the more favored Adamson ring. The radii should be $1\frac{1}{2}$ inches, and not less than $2\frac{1}{2}$ times the thickness of the plate.

6. *By the Adamson Ring.* This is generally considered the best joint for plain furnaces, but is more expensive than the angle or tee rings, although very much the stronger. Now that good flanging steel can be procured easily, it is not a difficult joint to make. The ends of the flue sheets are flanged out and are then

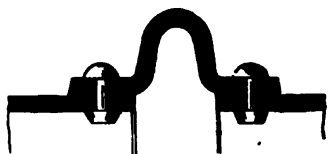


FIG. 43.—Flue Strengthening by the Bowling Hoop.

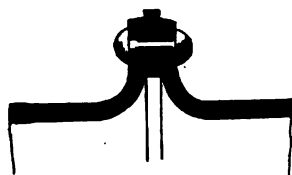


FIG. 44.—Flue Strengthening by the Adamson Ring.

riveted together through a ring inserted between them (Fig. 44). The object of the ring is not so much for strength as to facilitate calking the joint on the inside. Its thickness may be equal to that of the sheets. The flanging should be over a radius equal to $2\frac{1}{2}$ times the thickness of the sheets, so as to prevent grooving and permit of longitudinal expansion. The advantages are the facility for making and maintaining a tight joint; the double thickness, being in the water space, cannot be overheated; while the steam pressure tends to keep the joint closed. The objection is the difficulty of replacing the flue as ordinarily designed.

7. *By Galloway Tubes.* These tubes are built in the flues and are chiefly used to increase the heating surface (Fig. 45). Some-

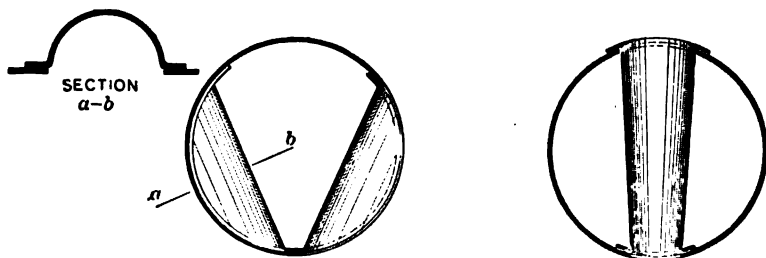


FIG. 45.—Flue Strengthening by Galloway Tubes.

times they are made as pockets on the side of the flue. They should be shaped like a truncated cone with the large end up, to facilitate circulation through them, and should be staggered so as

to assist in mixing the hot gases and thus promote combustion. They are generally about five or six inches in diameter at the small end, increasing to twice that at the large end (Figs. 19 and 20).

Large flues such as used in Cornish and Lancashire boilers do not rely on the additional strength lent by these tubes, but are also strengthened by rings of some convenient design. The disadvantage of these Galloway tubes is the trouble to clean and scale them.

8. *Flues* are made of other forms than plain, so as to be self-sustaining without the aid of stiffening rings. There are many patented shapes, but the principal ones in use are *Fox's Corrugated*

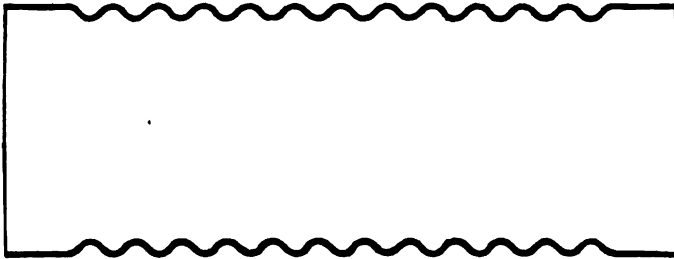


FIG. 46.—Fox's Corrugated Furnace Flue.

(Fig. 46), *Morison's Suspension* (Fig. 47), and *Purves' Ribbed Flues* (Fig. 48). These special flues are purchased from the makers, and can be obtained in various standard diameters, but at lengths to suit each order, and with either or both ends flanged as wanted. The illustrated catalogues of the makers should be consulted with

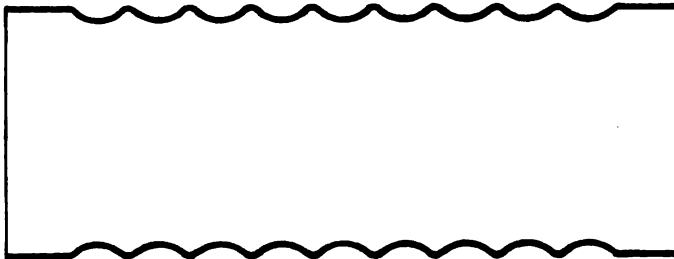


FIG. 47.—Morison's Suspension Furnace Flue.

respect to the various forms of flanging. Usually the flue ends are plain, and are single-riveted to flanges formed on the front head and

on the combustion-chamber. It is often convenient to make the front end of the flue $\frac{1}{4}$ or $\frac{3}{8}$ of an inch larger in outside diameter than the corrugations of the flue, so as to facilitate removal. The back end of the flue may be flanged to meet the combustion-chamber, and when so made it has the advantage of placing the rivet-heads out of the direct play of the flames, and in consequence will be more durable. This

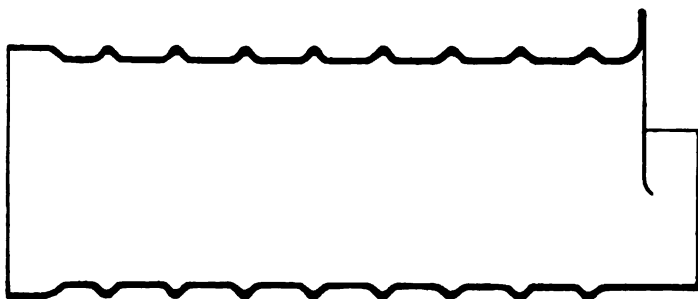


FIG. 48.—Purves' Ribbed Furnace Flue.

flange on the back end of the flue can be so shaped by the makers as to pass out through the hole in the front head. All these forms are good, and the disadvantage is the trouble to break off the scale from the corrugations. The flatter, therefore, the depressions, the greater will be the ease of cleaning, and the less the resistance offered to the passage of the gases. While the ribbed flues excel in this particular, they have the disadvantage of unequal thickness of metal at the ribs.

Reference is made to an article on "Marine Boiler Furnaces," by D. B. Morison, *Cassier's Magazine*, August, 1897. This article also gives the results of the experiments made by Otto Knaudt, of the firm of Schultz, Knaudt & Co., Essen, Germany, on the elasticity of form of furnace-flues.

Furnace-flues should not be tapped for the fastenings of bridge wall and grate bearers, as such tap bolts are very hard to keep tight. This is especially true for forms other than plain. Tapping is so easy that many builders unfortunately adopt this method.

All flues subjected to the direct heat of the fire should be designed so as to be as thin as possible. Many place the maximum thickness at $\frac{3}{4}$ -inch, but $\frac{1}{4}$ or $\frac{5}{8}$ -inch would be a better limit. Thick flues burn away very rapidly, until a thickness of

about $\frac{1}{8}$ is reached. Furthermore, thick furnaces are apt to become heated above the temperature at which the steel begins to lose in resisting strength.

The liners of steam-chimneys are flues and subject to the same arguments as stated above, except that they may be made thicker, because the gases have lost much of their heat before reaching them. Liners being superheating surfaces, should be proportioned with a larger factor of safety than that used for flues which are water-heating surfaces. They must not be made so thick as to be liable to burn.

Rules for Flues.

1. U. S. Board of Supervising Inspectors of Steam-vessels.

TUBES.

Lap-welded tubes, used in boilers whose construction was commenced after June 30, 1905, having a thickness of material according to their respective diameters, shall be allowed a working pressure as prescribed in the following table, provided they are deemed safe by the inspectors:

Outside diameter.	Thickness of material.	Greatest length allowable.	Maximum pressure allowable.
Inches.	Inch.	Feet.	Pounds.
1	.072	Any length	225
1 $\frac{1}{2}$.072	do.	225
1 $\frac{1}{2}$.088	do.	225
1 $\frac{3}{4}$.095	do.	225
2	.095	do.	225
2 $\frac{1}{2}$.095	do.	225
2 $\frac{1}{2}$.109	do.	225
2 $\frac{3}{4}$.109	do.	225
3	.109	do.	225
3 $\frac{1}{2}$.120	do.	225
3 $\frac{1}{2}$.120	do.	225
3 $\frac{3}{4}$.120	do.	225
4	.134	do.	225
4 $\frac{1}{2}$.134	do.	225
5	.148	do.	225
6	.165	do.	225

FLUES.

The annexed table shall include all such riveted and lap-welded flues exceeding 6 inches in diameter and not exceeding 40 inches in diameter not otherwise provided for by law.

TABLE OF STEAM PRESSURE PER SQUARE INCH ALLOWABLE ON RIVETED AND LAP-WELED FLUES MADE IN SECTIONS—Continued.

Thickness of material.	Greatest length of sections allowable, 30 inches.										Least thickness of material allowable.										Diameter of flues.									
	34 inch.	35 inch.	36 inch.	37 inch.	38 inch.	39 inch.	40 inch.	41 inch.	42 inch.	43 inch.	44 inch.	45 inch.	46 inch.	47 inch.	48 inch.	49 inch.	50 inch.													
Over 23, not over 24 inches.	Over 24 inches.	Over 25 inches.	Over 26 inches.	Over 27 inches.	Over 28 inches.	Over 29 inches.	Over 30 inches.	Over 31 inches.	Over 32 inches.	Over 33 inches.	Over 34 inches.	Over 35 inches.	Over 36 inches.	Over 37 inches.	Over 38 inches.	Over 39 inches.	Over 40 inches.													
Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.	Pounds pressure.
113	116	118	120	122	124	126	128	130	132	134	136	138	140	142	144	146	148	150	152	154	156	158	160	162	164	166	168	170	172	174
34-inch.	35-inch.	36-inch.	37-inch.	38-inch.	39-inch.	40-inch.	41-inch.	42-inch.	43-inch.	44-inch.	45-inch.	46-inch.	47-inch.	48-inch.	49-inch.	50-inch.	51-inch.	52-inch.	53-inch.	54-inch.	55-inch.	56-inch.	57-inch.	58-inch.	59-inch.	60-inch.	61-inch.	62-inch.	63-inch.	64-inch.

To find steam pressure allowed on flues in the above table, multiply crushing stress (8,000 lbs.) by thickness of material and divide the product by diameter of flue.

For any such flue requiring more pressure than is given in table, the same will be determined by proportion of thickness to any given pressure in table to thickness for pressure required, as per example:

A flue not over 19 inches in diameter and 3 feet long requires a thickness of .39 of an inch for 164 pounds pressure; what thickness would be required for 250 pounds pressure?

$$164 : 250 :: .39 : .5946,$$

or a thickness of .595 inch.

Or, if .39 inch thickness gives a pressure of 164 pounds, what will .595 inch thickness give?

$$.39 : .595 :: 164 : 250 \text{ pounds required.}$$

And all such flues shall be made in sections, according to their respective diameters, not to exceed the lengths prescribed in the table, and such sections shall be properly fitted one into the other and substantially riveted, and the thickness of material required for any such flue of a given diameter shall in no case be less than the least thickness prescribed in the table for any such given diameter; and all such flues may be allowed the prescribed working steam pressure if, in the opinion of the inspectors, it is deemed safe to make such allowance. Inspectors are therefore required, from actual measurement of each flue, to make such reduction from the prescribed working steam pressure for any material deviation in the uniformity of the thickness of material, or from any material deviation in the form of the flue from that of a true circle, as in their judgment the safety of navigation may require.

FURNACES.

The tensile strength of steel used in constructing furnaces shall not exceed 67,000, and be not less than 58,000 pounds. The minimum elongation in 8 inches shall be 20 per cent.

All corrugated furnaces having plain parts at the ends not exceeding 9 inches in length (except flues especially provided for), when new, and made to practically true circles, shall be allowed a steam pressure in accordance with the following formula:

$$P = \frac{C \times T}{D}$$

LEEDS SUSPENSION BULB FURNACE.

$$P = \frac{C \times T}{D}$$

where P = pressure in pounds ;

T = thickness in inches, not less than five-sixteenths of an inch ;

D = mean diameter in inches ;

C = a constant, 17,300, determined from an actual destructive test under the supervision of the Board, when corrugations are not more than 8 inches from center to center, and not less than $2\frac{1}{2}$ inches deep.

MORISON CORRUGATED TYPE.

$$P = \frac{C \times T}{D}$$

where P = pressure in pounds ;

T = thickness in inches, not less than five-sixteenths of an inch ;

D = mean diameter in inches ;

C = 15,800, a constant, determined from an actual destructive test under the supervision of the Board of Supervising Inspectors, when corrugations are not more than 8 inches from center to center, and the radius of the outer corrugations is not more than one-half of the suspension curve.

[In calculating the mean diameter of the Morison furnace, the least inside diameter plus 2 inches may be taken as the mean diameter, thus :

Mean diameter = least inside diameter + 2 inches.]

FOX TYPE.

$$P = \frac{C \times T}{D}$$

where P = pressure in pounds ;

T = thickness in inches, not less than five-sixteenths ;

D = mean diameter in inches ;

C = 14,000, a constant, when corrugations are not more than 8 inches from center to center and not less than $1\frac{1}{2}$ inches deep.

PURVES TYPE.

$$P = \frac{C \times T}{D}$$

where P = pressure in pounds ;

T = thickness in inches, not less than seven-sixteenths ;

D = least outside diameter in inches ;

$C = 14,000$, a constant, when rib projections are not more than 9 inches from center to center and not less than $1\frac{1}{8}$ inches deep.

The thickness of corrugated and ribbed furnaces shall be ascertained by actual measurement. The manufacturer shall have said furnace drilled for a one-fourth inch pipe tap and fitted with a screw plug that can be removed by the inspector when taking this measurement. For the Brown and Purves furnaces the holes shall be in the center of the second flat ; for the Morison, Fox, and other similar types in the center of the top corrugation, at least as far in as the fourth corrugation from the end of the furnace.

TYPE HAVING SECTIONS 18 INCHES LONG.

$$P = \frac{C \times T}{D}$$

where P = pressure in pounds ;

T = thickness in inches, not less than seven-sixteenths ;

D = mean diameter in inches ;

$C = 10,000$, a constant, when corrugated by sections not more than 18 inches from center to center and not less than $2\frac{1}{4}$ inches deep, measuring from the least inside to the greatest outside diameter of the corrugations, and having the ends fitted one into the other and substantially riveted together, provided that the plain parts at the ends do not exceed 12 inches in length.

ADAMSON TYPE.

When plain horizontal flues are made in sections not less than 18 inches in length, and not less than five-sixteenths of an inch thick, and flanged to a depth of not less than three times the diameter of rivet-hole plus the radius at furnace wall (inside diameter of furnace), the thickness of the flanges to be as near the thickness of the body of the plate as practicable.

The radii of the flanges on the fire side shall be not less than three times the thickness of plate.

The distance from the edge of the rivet-hole to the edge of the flange shall be not less than the diameter of the rivet-hole, and the diameter of the rivets before driven shall be at least one-fourth inch larger than the thickness of the plate.

The depth of the ring between the flanges shall be not less than three times the diameter of the rivet-holes, and the ring shall be substantially riveted to the flanges. The fire edge of the ring shall terminate at or about the point of tangency to the curve of the flange, and the thickness of the ring shall be not less than one-half inch.

The pressure allowed shall be determined by the following formula :

PLAIN CIRCULAR FURNACES OR FLUES, AND ADAMSON FURNACES MADE IN SECTIONS NOT LESS THAN 18 INCHES IN LENGTH.

$$P = \frac{51.5}{D} [18.75T - (L \times 1.03)]$$

where P = working pressure in pounds per square inch ;

D = outside diameter of furnace in inches ;

L = length of furnace in inches ;

T = thickness of plate in sixteenths of an inch.

VERTICAL TYPE.

Cylindrical flues used as furnaces in vertical boilers, when new, and made to practically true circles, shall be allowed a steam pressure by the following formula :

$$P = \frac{C \times T}{D}$$

where P = pressure of steam allowable in pounds ;

T = thickness of flue in inches, not less than one-fourth ;

D = outside diameter of flue in inches, not to exceed 42 inches ;

C = 10,577, a constant, when the length of the flue does not exceed 42 inches, measuring from the center of the rivet-holes in the head to the center of the rivet-holes in the-leg.

When the flue exceeds 42 inches in diameter, it is deemed to be flat surface and must be stayed accordingly.

STEAM-CHIMNEY FLUES.

The Morison, Fox, Purves, or Brown types of corrugated furnaces may be used as flues for steam chimneys or superheaters and shall be allowed a steam pressure by their respective formulæ, and other flues, as described below, when new and made to practically true circles, shall be allowed a steam pressure by the following formula :

$$P = \frac{C \times T}{D}$$

where P —pressure in pounds ;

T —thickness of material in inches ;

D —outside diameter of flue in inches ;

$C=12,000$ for flues under 30 inches in diameter, plates at least five-sixteenths of an inch thick, supported by angle rings at least $2\frac{1}{2}$ by $2\frac{1}{2}$ inches ;

$C=12,000$ for flues 30 inches and under 45 inches in diameter, plates at least three-eighths of an inch thick, supported by angle rings at least $2\frac{1}{2}$ by $2\frac{1}{2}$ inches ;

$C=12,000$ for flues 45 inches and under 55 inches in diameter, plates at least seven-sixteenths of an inch thick, supported by angle rings at least 3 by 3 inches ;

$C=12,000$ for flues 55 inches and under 65 inches in diameter, plates at least one-half inch thick, supported by angle rings at least 3 by 3 inches ;

$C=12,000$ for flues 65 inches and under 75 inches in diameter, plates at least nine-sixteenths of an inch thick, supported by angle rings at least $3\frac{1}{2}$ by $3\frac{1}{2}$ inches.

$C=12,000$ for flues 75 inches and under 85 inches in diameter, plates at least five-eighths of an inch thick, supported by angle rings at least $3\frac{1}{2}$ by $3\frac{1}{2}$ inches.

$C=12,000$ for flues 85 inches in diameter, plates at least eleven-sixteenths of an inch thick, supported by angle rings at least 4 by 4 inches.

For flues over 85 inches in diameter, add one-sixteenth of an inch to eleven-sixteenths of an inch for every 10 inches increase in the diameter of the flue.

The distance, center to center, between angle rings, or center of angle rings to center of rivets in the heads, shall in no case exceed $2\frac{1}{2}$ feet.

The angle rings shall be accurately fitted and substantially riveted to the flue and connected to the outer shell by braces, which braces shall not exceed 30 inches from center to center on the flue.

2. Board of Trade (British).

The working pressure in pounds per square inch is

$$= \frac{\text{Constant} \times \text{square of thickness of plate in inches}}{(\text{Length in feet} + 1) \times \text{diameter in inches}},$$

provided that the pressure does not exceed $\frac{9000 \times \text{thickness in inches}}{\text{diameter in inches}}.$

Value of Constants:

Furnaces with butt- joints and drilled rivet-holes.	{	90,000 where the longitudinal seams are welded.
		90,000 where the longitudinal seams are double-riveted and fitted with single butt-straps.
		80,000 where the longitudinal seams are single-riveted and fitted with single butt-straps.
		90,000 where the longitudinal seams are single-riveted and fitted with double butt-straps.
Furnaces with butt- joints and punched rivet-holes.	{	85,000 where the longitudinal seams are double-riveted and fitted with single butt-straps.
		75,000 where the longitudinal seams are single-riveted and fitted with single butt-straps.
		85,000 where the longitudinal seams are single-riveted and fitted with double butt-straps.
Furnaces with lapped joints and drilled rivet-holes.	{	80,000 where the longitudinal seams are double-riveted and bevelled.
		75,000 where the longitudinal seams are double-riveted and not bevelled.
		70,000 where the longitudinal seams are single-riveted and bevelled.
		65,000 where the longitudinal seams are single-riveted and not bevelled.
Furnaces with lapped joints and punched rivet-holes.	{	75,000 where the longitudinal seams are double-riveted and bevelled.
		70,000 where the longitudinal seams are double-riveted and not bevelled.
		65,000 where the longitudinal seams are single-riveted and bevelled.
		60,000 where the longitudinal seams are single-riveted and not bevelled.

The above constants are for use when flues are made of iron; but when of steel, increase the constants 10 per cent. The length is to be measured between the rings if the furnace is made with rings.

When furnaces are machine-made, of the Fox corrugated, the Morison suspension, or the Purves ribbed types, and the plates are not less than $\frac{1}{4}$ inch thick, the working pressure per square inch

$$= \frac{14,000 \times \text{thickness in inches}}{\text{Outside diameter in inches}}$$

The diameter is measured at the bottom of the corrugations or over the plain part between ribs.

When furnaces of ordinary diameter are constructed of a series of rings welded longitudinally, and the ends of each ring flanged and the rings riveted together, and so forming the furnace, the working pressure is found by the following formula, provided the length in inches between the centres of the flanges of the rings is not greater than $(120t-12)$, and the flanging is performed at one heat by machine:

$$\frac{9900 \times t}{3 \times d} \left(5 - \frac{l+12}{60 \times t} \right),$$

in which t denotes thickness of plate in inches;

l " length between centre of flanges in inches;

d " outside diameter of furnace in inches.

The radii of the flanges on the fire side should be about $1\frac{1}{2}$ inches. The depth of the flanges from fire side should be three times the diameter of the rivet plus $1\frac{1}{2}$ inches, and the thickness of the flanges should be as near the thickness of body of plates as practicable. The distance from edge of rivet holes to edge of flange should not be less than diameter of rivet, and the diameter of rivet at least $\frac{3}{4}$ inch greater than the thickness of plate. The depth of ring between flanges should be not less than three times the diameter of rivet, the fire edge of ring should be at about the termination of the curve of flange, and the thickness not less than half the thickness of the furnace plate. It is very desirable that these rings should be turned. After all welding, flanging and heating is completed each ring should be efficiently annealed in one operation.

Tubes. In fire-tubular boilers, the number of tubes and the size required depend on the area through them for purposes of draft and on the amount of heating surface necessary to absorb the heat of the fire. The area for draft has been discussed in a previous chapter.

With regard to heating surface, it must be remembered that increasing the diameter increases the surface in direct proportion, and the calorimeter in proportion to the square of the diameters. The size needed to suit both calorimeter and heating surface can be determined by trial until one size is found which will give a result nearest to that wanted.

The size is also generally dependent on the quality of fuel to be used. With hard coals, tubes 3 inches and less in diameter are used; while with soft coals, 3 inches and over are preferred, as smaller tubes choke up too rapidly with soot.

It is not advisable to make tubes more than 50 or 60 diameters in length, as the additional length loses rapidly in evaporating efficiency. Instead of increasing the length, it is better to use more tubes of smaller diameter, even though the proposed calorimeter cannot be obtained, provided, of course, that too small a calorimeter is not used so as to restrict the draft.

In large boilers it is best to arrange the tubes in banks or nests, by leaving out the vertical middle row. This wide water space assists the convection currents and, therefore, the evaporative power of the boiler. In Scotch boilers, the tubes always should be thus divided into nests.

The outer rows also should be kept far enough away from the shell, so as not to interfere with the downward currents; but since the tubes act as stays they must not be placed too far from the shell, or the tube plates will have to be made too thick. In horizontal return-tubular and similar boilers the distance between outer tubes and shell should not be less than 3 inches, and when the boilers exceed 50 inches diameter it should be more.

The tubes should be arranged in horizontal and vertical rows, so that the steam bubbles can have a direct passage through the vertical spaces to the surface, and not be staggered except in locomotives. The pitch of the tubes horizontally should never be less than that of the vertical rows. When the pitch vertically is small, then the horizontal pitch should be greater by at least an eighth to a quarter inch.

The pitch ought not to be less than 1.4 times the diameter under ordinary conditions, but if care be taken in the general design, and the tubes are not over 30 diameters in length, they can be spaced nearer together. Three-inch and smaller tubes could be spaced so as to leave only $\frac{3}{4}$ inch between holes in tube plates, but such small pitch is too close for good steaming purposes. The distance between such tubes, if possible, ought not to be less than 1 inch vertically and $1\frac{1}{2}$ inch horizontally. If proper spacing be neglected, priming is apt to occur when the boiler is forced.

The top row of tubes should be placed low enough so as to leave ample steam space and permit some of the water surface to extend over the spaces left for downward currents at the sides, between the tubes and the shell. As a general thing it is difficult to get the tubes low enough.

In large shell boilers, like the Scotch, the top row should not be

higher than one-third the diameter of shell from the top, but may be 0.28 of the diameter when the outflow of steam is very regular, as for marine engines of comparatively short stroke. In horizontal return-tubular boilers the uppermost row is generally about two-fifths of shell diameter from top.

The best method is to make preliminary sketches, to scale, of the proposed cross-section, and locate by trial and error the positions of tubes, flues, stays and other internal arrangements.

Provided that the total area of tube opening be neither too large nor too small, the economy appears to be little affected by the size of the tube employed. Still, as small tubes become more rapidly choked with soot than large ones, the latter had better be used with soft coals and wood, unless the conditions are such as to permit frequent cleaning. Tube diameters are always given on outside measurement, so that the area of opening corresponds to a diameter equal to that of tube less twice the thickness.

The tubes act as stays in supporting the tube plates. In large boilers with high steam pressures, some of the ordinary tubes are replaced with extra heavy tubes, called "stay tubes." The use of these stay tubes is more common in European practice than in this country, and more common in marine than in stationary boilers. When used, every fourth tube is generally made a stay tube.

Boiler tubes are made of solid drawn brass, and of lap-welded iron or steel. Steel tubes can also be made solid drawn. Brass tubes are little used at present, preference being given to charcoal iron or to mild steel. Steel tubes are said to be less durable than iron tubes, but the fault is chiefly in the poor quality of steel furnished. The best steel should be just as able to withstand wear and corrosion as the best iron. Poor qualities are more difficult to detect by visual inspection in steel than in iron, and for cheap work iron tubes, therefore, are preferred by many.

Nickel-steel has been used to some extent for boiler-tubes, and the result so far has been favorable. A. F. Yarrow* made some experiments on nickel versus mild steel for water tubes with the following results: The nickel alloy contained from 20 per cent to 25 per cent of nickel. Boiler tubes deteriorate from three principal causes, (a) action of acids, due to grease; (b) overheating and

* Inst. of Naval Architects, July, 1899, and Journal American Society Naval Engineers, August, 1899.

oxidizing through contact with hot gases; (c) action of superheated steam, which decomposes.

The corrosion tests showed that mild steel lost sixteen and a half times more weight than nickel-steel, and the oxidization tests two and nine-tenths times. The superheated steam tests showed a loss in nickel-steel of 12.7 grammes against 85.2 grammes in mild steel, or that mild steel tubes would have to be replaced two and one-third times as often as nickel-steel ones. The expansion test showed that nickel-steel expands more than mild steel in the ratio of four to three. Small amounts of nickel, about five per cent, produced only slight gains.

TABLE XVII

THE GREATEST NUMBER OF TUBES USUALLY PLACED IN HORIZONTAL RETURN TUBULAR BOILERS

Diameter of Boiler-Shell in Inches.	Diameter of Tubes.		
	3 Inches.	3½ Inches.	4 Inches.
30	20	14	10
36	32	22	16
42	45	32	25
44	48	33	26
48	56	38	27
50	60	40	30
54	70	48	36
60	84	62	48
62	90	70	52
64	96	78	58
66	104	86	64
72	126	98	78
78	160	124	100

The holes in the tube plates should be drilled to a perfect fit, and the burr should be removed with light filing. The tubes should be cut to proper length and fitted into place. The tube makers will supply the tubes of the desired length, as well as slightly upsetting one end to $\frac{1}{16}$ larger diameter. This enlargement of the end facilitates the tube's removal at any time, but is a refinement seldom adopted, except for stay tubes or for tubes screwed into tube sheets.

Plain tubes are fastened to the tube sheets by expanding the ends. This is done by a special tool called a "tube expander," which operates by simply stretching the metal so as to closely fit the hole. If a tube leaks at the joint, it may be expanded again.

There are two forms of expanders in common use: The Prosser expander (Fig. 49) consisting of a tapered mandril carrying segments

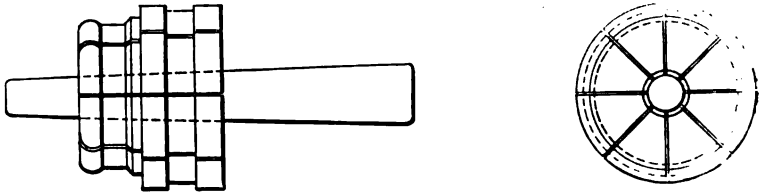


FIG. 49.—Prosser's Tube Expander.

with radial joints, and operated by repeatedly driving in the mandril, slightly turning the segments at each operation; and the

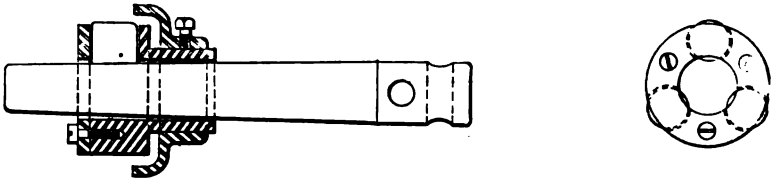


FIG. 50.—Dudgeon's Tube Expander.

Dudgeon expander (Fig. 50) consisting of a tapered mandrel carrying a hollow steel head with rollers, and operated by turning the

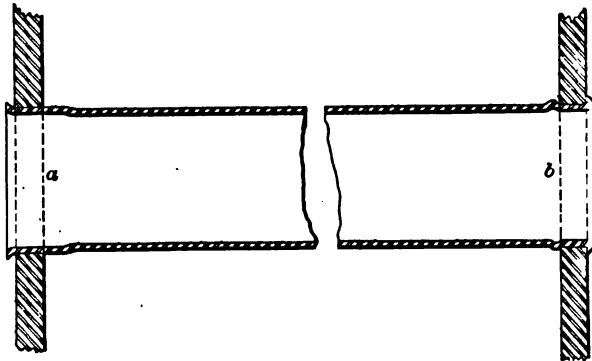


FIG. 51.—Effect of Expanding the Tube Ends. *a*. Expanded by Dudgeon's expander and flared. *b*. Expanded by Prosser's expander and beaded.

mandrel while it is gently forced in, thus rolling the tube metal out into a tight fit. The effect of expanding the tube ends is shown in Fig. 51.

Of the two types, the latter is less apt to cause injury to the tube

end or to distort the tube-sheet when used by inexperienced men, and is much liked on account of its rolling action and the ease with which it can be adjusted to suit varying thickness of tube-sheets.

Tubes exceeding 5 inches in diameter cannot be expanded so as to form a permanent tight joint, as the metal stretches away at one place while being pressed tight at another.

The projecting tube ends may be beaded over—that is, rolled back outwardly so as to cover the joint (Fig. 51*b*). This makes a very neat appearance when well done, and is the common American practice. Only tubes of good, soft material will bead smoothly, and tubes that show fraying at the bead had better be cut out and replaced. Probably as good a practice as any is, after expanding, to bead the tube at the fire-entering end, and to drive a slightly tapered plug into the fire-exit end, so as to make it slightly flared (Fig. 51*a*). Afterwards these ends should be all milled off evenly and not leave over $\frac{3}{16}$ -inch to project beyond the tube-sheet.

Some engineers prefer to make the hole in the tube-sheet tapered, to increase the holding power of the tube. If so made, the inner sharp edge must be slightly rounded. The holding power of expanded tubes, even if not beaded or wedged out like a cone, has proved to be ample to support the tube-sheets.

Stay-tubes are generally made $\frac{1}{4}$ -inch in thickness for all pressures, but are sometimes made slightly heavier. They are fastened by screwing into the tube-plates, and sometimes carry nuts on the outside. Stay-tubes have not been found necessary except for very heavy pressures and wide spacing. One end should be upset so as to easily set and withdraw them, and the threads on both ends should be continuous.

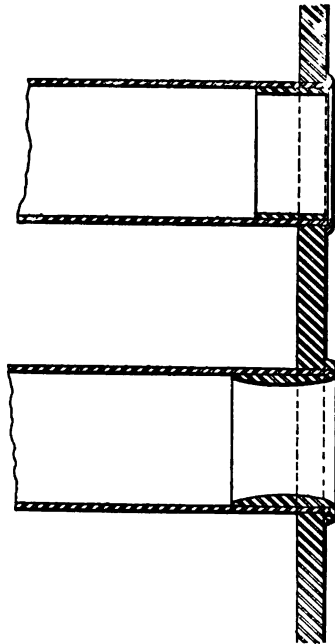


FIG. 52.—Ferrules for Tube Ends.

In order to increase the holding power, to prevent the ends from becoming overheated and from leaking, many engineers drive ferrules (Fig. 52) into the tubes. These ferrules are made of iron or copper rings, or of malleable cast iron in the shape of a nozzle. They are generally put in the fire-entering end, although many locomotives have them in the front or exit end. They often stop leaking tubes by keeping the tube end cool. Leaking tubes are frequent in some boilers. The trouble may be caused by expansion through too stiff a tube-plate. These tube-plates should be as thin as can be conveniently made.

Retarders (Fig. 53) are often placed in the tubes in order to more thoroughly mix the products of combustion and to break up the "steam lines," so that all hot particles will come into contact with the tube surface. These retarders are frequently of patented forms, but more or less resemble a spiral or corkscrew shaped piece of metal set inside the tube. They can be withdrawn for cleaning the tube, or replaced when worn out. They work best with a very strong natural or mechanical draft. Tubes with retarders should be one size larger than those without, under similar conditions.

Retarders have proved beneficial with the use of fuel-oil, as the products of combustion are apt to pass too freely through the tubes of the boiler and escape into the stack, with the result that the water does not absorb the heat from the oil vaporized in the furnace. This difficulty is especially apparent in boilers designed for using either coal or oil as a fuel; and the cause can be attributed to the fact that the tubes, when made sufficiently large to allow an accumulation of soot without obstructing the draft, have too great a sectional area when oil is used. With oil fuel there is practically no soot to collect, and the hot gases rush through them under the impetus given



Fig. 53.—Retarder for Tubes.

by the pressure of the burner and the strong draft produced by the stack.

Luther D. Lovekin has invented the Acme patent retarder shown in Fig. 54, made of refractory clay and in shape somewhat like the Admiralty ferrule. These retarders are inserted at each end of the tubes, causing a reduction in area, and consequently retard the passage of the hot gases. The refractory nature of

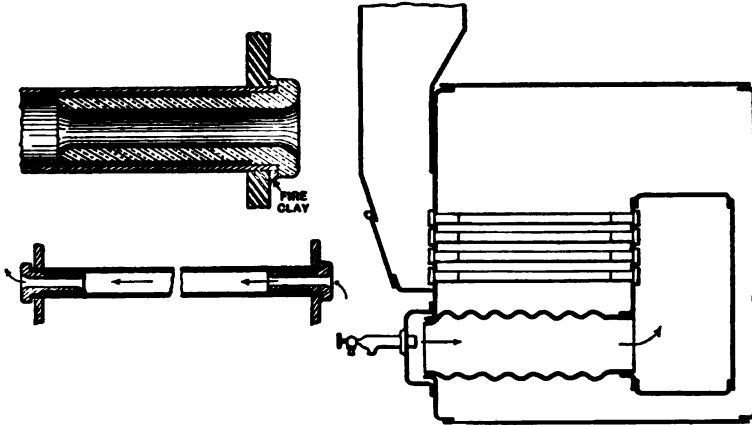


FIG. 54.—Acme Refractory Clay Retarder for use with Fuel Oils.

these Acme retarders will cause them to retain their incandescence for some time after the fuel has been shut off and will tend to reignite the gases should the flame become accidentally extinguished by water getting into the oil.

The Acme retarders have been tried on the steamships "Ligonier" and "Larimer" of the Guffey Petroleum Company, 1903. They produced no smoke and were found to reduce the temperatures, when used with the Lassöe-Lovekin air-blast and Rockwell steam-blast pulverizers, as follows:

	Temperature at Base of Stack.
Trial without any retarders.	850° F.
" with spiral steel retarders	750° "
" with spiral steel retarder, and with Acme retarder at front end of tube only.	680° "
" with Acme retarders at both ends of tubes (no spiral steel retarder)	550° "
With smaller opening in retarders the temperature could be as low as	450° "

The Serve tube (Fig. 55) is a tube having internal ribs reaching about half-way to the centre. These ribs increase the heat

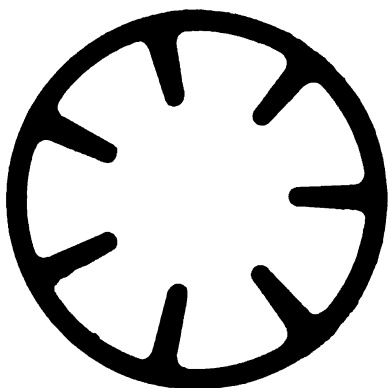


FIG. 55.—Serve Tube.

transmitting power of the tube, and effect some economy. It is much stiffer and much more durable than the ordinary tube. It is more costly at first, but probably, in many cases, cheaper in the end. The ribs interfere with cleaning to some extent. As some area is occupied by the ribs, one size larger than plain tubes should be used in order to obtain the required calorimeter.

Tubes are made of standard or list thickness, but can be ordered of any thickness as required.

RULES FOR THICKNESS OF BOILER-TUBES.

1. U. S. Board of Supervising Inspectors of Steam-vessels. Given under the rules for flues.

2. As given by A. E. Seaton in "Manual of Marine Engineering," for minimum thickness in numbers of Birmingham Wire Gauge.

External diameter of tubes.....	2	2½	2½	2½	3	3½	3½	3½	4
List or thickness for 40 lbs.....	12	12	11	11	11	10	10	10	9
Thickness for under 90 lbs.....	11	11	10	10	10	9	9	9	8
Thickness for over 90 lbs.....	11	10	9	9	9	8	8	8	7

Stays. Parts of many boilers must be stayed in order to withstand the steam pressure. Flat surfaces obtain their resisting strength from the stays, and the thickness of flat plates is dependent on the distance between stays. There is, therefore, a choice for the designer in determining for each case the proper distance between stay centres and the requisite plate thickness, or vice versa.

The tubes act as stays, as has already been seen, and give the required support to the tube-sheets.

Stays may be of round, square or flat sections, as most convenient.

They may be passed through the stayed sheet with the ends riveted over, or screwed into the sheet with riveted ends, or screwed into the sheet with a nut or with a nut and a washer. Flat or square stays are usually flanged and forged on the end to

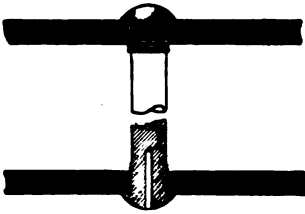


FIG. 56.—Screw Stay, ends upset and riveted.

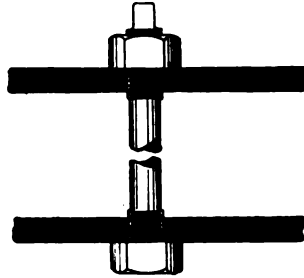


FIG. 57.—Screw Stay, ends upset and fitted with nuts.

produce a palm for riveting, or may have a single or double palm end forged on. Sometimes a tee or angle is riveted to the stayed sheet, and the stay fastened to such piece or pieces by a bolt or bolts. In many instances gusset plates are used instead of stay-rods. When the area of flat surface to be supported is small, it often can be stiffened sufficiently by simply riveting a tee or two angles back to back.

When screw-stays are used the ends should be “upset” (Figs. 56, 57, and 61), so as to retain the full strength at the bottom of the

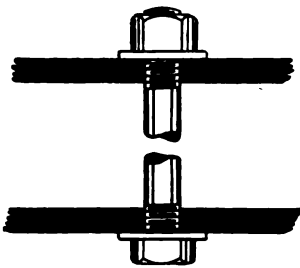


FIG. 58.—Screw Stay, ends not upset, fitted with nuts and washers.



FIG. 59.—Stay fitted with ferrule.

threads. Short screw-stays, such as are used to stay water-legs and similar places, often can be conveniently made by cutting the thread on the full length and then turning off the thread on the part

between the sheets. This will leave the twist of the thread continuous and facilitate its insertion. A smooth surface is better able to withstand corrosion, which is very liable to attack the metal at the base of the thread.

Small screw-stays and stay-bolts can be fastened by riveting over the ends of the stays (Figs. 56 and 59), but this method should not be adopted when the thickness of sheet is less than half the diameter of stay. In such cases it is best to fasten with a nut on the outside end of stay. (Figs. 57 and 58.)

Stays that screw into the sheet make a tighter and more durable joint than those which are simply passed through a plain hole. Screw-stays always should be used in high-pressure boilers. Screw stay-bolts may be calked to drive the metal into the threads, which is good practice. Stay-bolts that are not screwed into the sheets should be passed through a ferrule or distance-piece (Fig. 59) to prevent the sheet from being bent inward, when stay end is riveted over or the nuts are set up. If the ferrules are omitted, some stays may carry a greater load than adjacent ones. The great objection is the difficulty of getting all the ferrules of even length, while many claim as an advantage that the ferrules protect the stay from corrosion.

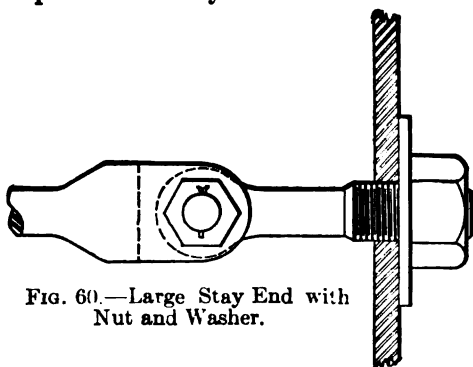


FIG. 60.—Large Stay End with Nut and Washer.

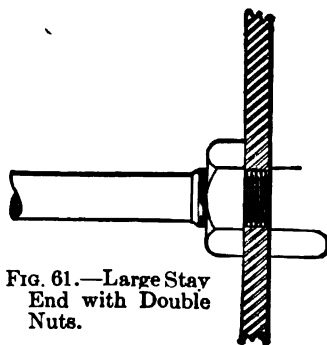


FIG. 61.—Large Stay End with Double Nuts.

These small stays often are drilled with a hole about $\frac{1}{8}$ inch in diameter along the axial line to act as a tell-tale (Fig. 56). Should corrosion eat through the stay, steam will blow out of this hole. It is common practice in locomotive work to drill a tell-tale in every second stay of the water-legs of fire-box. These tell-tale holes reduce the ability of the stay to withstand repeated bending due to the expansion of the sheets. See paper by F. J. Cole, Trans. Am. Soc. M. E., June 1898.

All large stays passing through the sheets should be fastened with a nut, and when necessary with a washer. For very large stays, or for stays sustaining heavy pressures, an additional nut or lock-nut on the inside should be used (Fig. 61). When double nuts are used, the stay may or may not be screwed into the plate, but a much better and tighter joint is made by utilizing the thread. The nuts in all cases should have a shallow groove turned on the side next to the plate in order to hold a packing (Fig. 62), usually made of asbestos or cotton waste and red lead, or of cement.

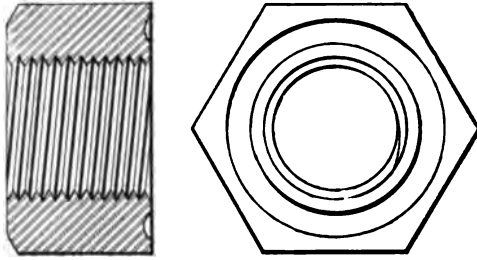


FIG. 62.—Nut for Stays, showing Packing Groove.

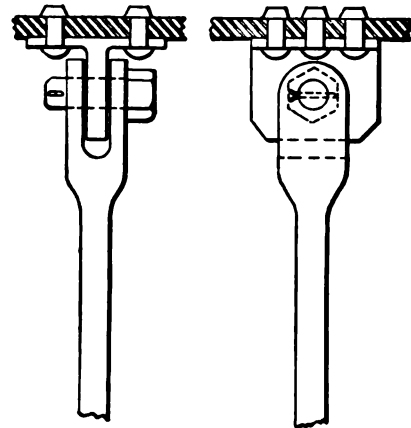


FIG. 63.—Stay End with Bolt in Double Shear.

attachments in tension, which is when ample rivet section is provided.

Where this is not done, these large stays often have to be cut and removed through the manhole in pieces. Such stays when cut can be re-united by upsetting the ends and fastening with a heavy coupling or turn-buckle, but such a method cannot be recommended except for emergencies.

Large stays can be fastened by pins, bolts or keys to angles or tees riveted to the flat sheets. This method places the rivets of the end objectionable, but allowable. Such rivets should not carry

over 6000 pounds per square inch of sectional area. The bolts, etc., should always be arranged for double shear, either by having double eyes on the stay end to straddle the leg of the tee (Fig. 63), or better by using two angles with the stay inserted between them (Fig. 64). Sometimes a tee end is forged on the stay end, so as to carry a number of bolts in order to furnish sufficient bearing surface to make them equal in strength to the body of the stay.

When pins are used, cotters should be passed through the ends (Figs. 63 and 64) so as to prevent their falling out. If bolts are used, then double or lock nuts are to be preferred to single nuts, and the bolt so placed as not to fall out by gravity should the nuts be acci-

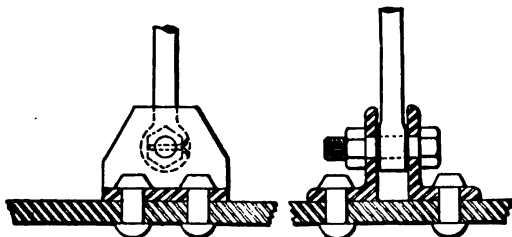


FIG. 64.—Stay End with Bolt in Double Shear.

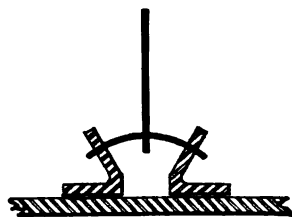


FIG. 65.—A Method of Failure of Fig. 64.

dentally removed. If neither cotters nor nuts be used, the angles often spread apart, and the pin bends under the increased leverage, allowing the stay to become slack (Fig. 65).

When the area to be supported is small, sufficient strength is often secured by riveting on the plate a tee or double angles back to back. The angles are to be preferred, and are best arranged radially if possible. The rivets should be spaced as if stay-bolts. These ribs should, however, be limited to small areas that cannot be otherwise stayed, or to boilers designed for very low pressures.

Gusset-plates are often used, with the advantage that they act over large areas and give great stiffness. Gussets should always be joined by double angles with the rivets in double shear (Fig. 66). Double gussets and single tees are not to be relied upon, since it is practically impossible to make the two gussets exactly alike in shape and material, so as to evenly divide the load to be supported. The gussets should be cut away so as not to come too close to the corner

between the head and the shell.

In general the gusset should not be nearer than four inches, and the greater the distance the better, in order that the head may have sufficient "play" to prevent grooving.

Other forms of stay ends are shown in Figs. 67, 68, 69 and 70.

Crown-sheets of fire-boxes are stayed either directly to the shell or by stay-bolts to a girder. In the former plan each stay is made as nearly perpendicular to the crown-sheet as possible. In consequence the stays often meet the shell at very acute angles, which is objectionable. In order to avoid this, the main boiler-shell over the fire-box can be made flat, and be carried parallel to the crown-

FIG. 66.—Gusset Plate Stay.

sheet, as in the "Belpaire" fire-box. This makes a large throat-sheet (the sheet joining the fire-box end to the cylindrical part) which is objected to by many engineers as being the weakest sheet in the shell.

Direct staying of the crown makes the best arrangement for strength, but fills up the boiler with stays and thus renders inspection, cleaning and scaling difficult.

The girders (Figs. 71 and 72) reach across the crown-sheet by resting on the end plates of the fire-box, forming a sort of bridge. From these

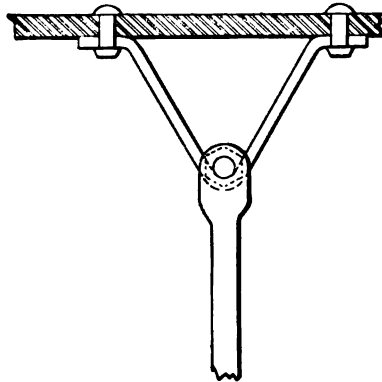


FIG. 67.—Stay End Fitted to Stirrup to Distribute the Support.

girders stay-bolts support the crown-sheet. The bottom of these girders should be sufficiently high above the crown-sheet (say

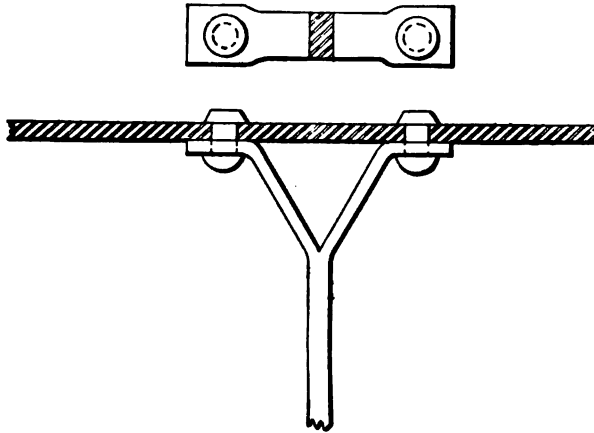


FIG. 68.—Stay End Split to Form Stirrup to Distribute the Support.

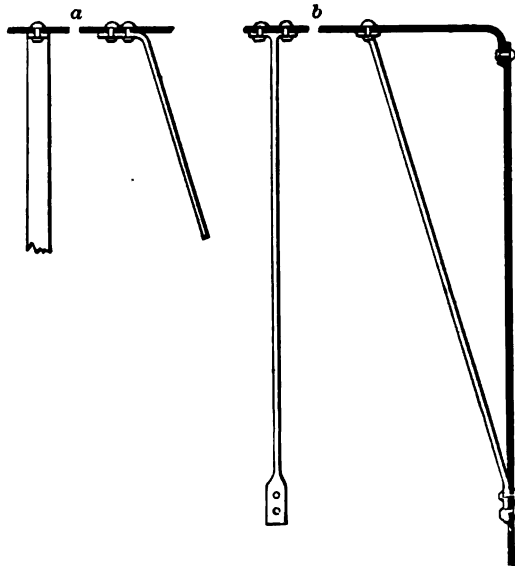


FIG. 69.—Diagonal Stay with rivets through palm in tandem at *a*, and in parallel at *b*. The form at *b* considered the stronger.

not less than $1\frac{1}{2}$ inches, depending on the design) to enable all scale and sediment to be readily removed. The stay-bolts should have ferrules.

Great care must be taken to rest the girder ends firmly on the end plating and to see that the pressure is not sufficient to crush

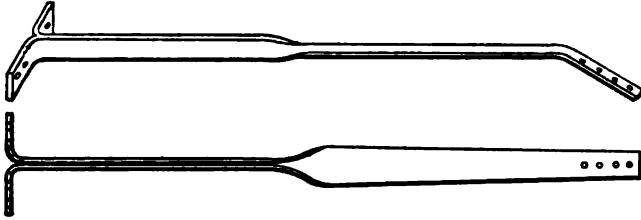


FIG. 70.—Huston Form of Stay without Weld.

those plates. Girders may be strengthened by having stays fastened to the shell. In Fig. 72 three forms of such stays are shown.

The girders can be made of forgings of steel or iron, or of cast steel, having holes for the stays (Fig. 71); or of two pieces, side by side, with the stays between (Fig. 72). The stay ends are then supported by a distance-washer, and the two half-girders riveted or bolted together through spacing-pieces or ferrules.

Stays are a necessary evil in all boilers; and, as they are apt to give trouble, the greatest care should be exercised in their design

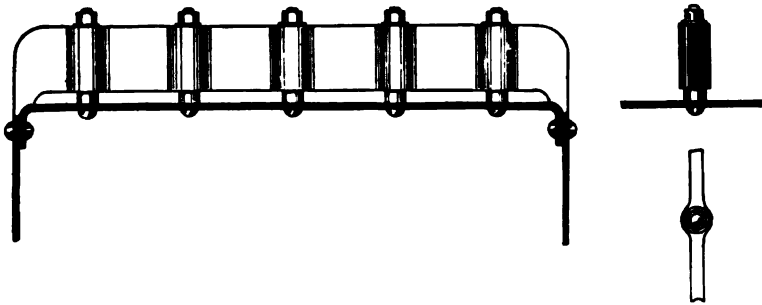


FIG. 71.—Girder Stay for Supporting Crown-sheet.

and spacing. They should be arranged so as not to obstruct the operation of cleaning and scaling. In the steam-space they should be arranged so that a man can pass between them when they are through stays from head to head; that is, be about 14 inches centre to centre, and be placed in horizontal and vertical rows.

Small stays, in narrow spaces like water-legs, must be arranged to permit a cleaning-tool to be inserted. These water-legs should be made as wide as convenient to suit the design. The inner sheet, exposed to the direct heat of the fire, will expand more than the

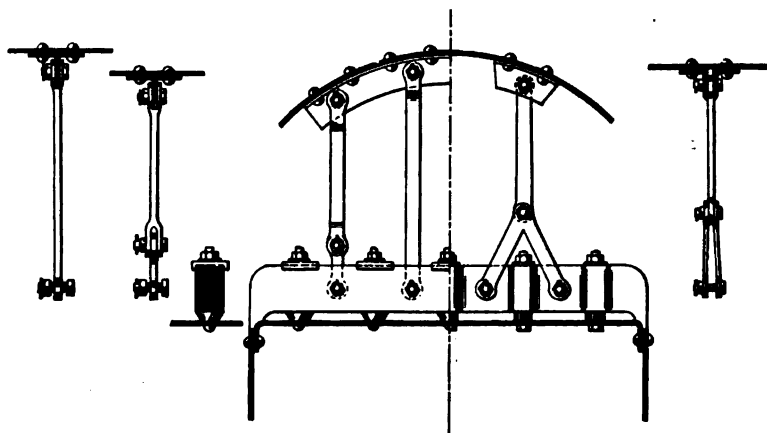


FIG. 72.—Girder Stay for Supporting Crown-sheet, showing three forms of strengthening stays to shell.

outer shell, and is liable to cause rupture of stay-bolts due to a repeated bending action. As the amount of bending is constant, the actual bending of short stays will be greater than that of long ones, and the former will therefore fail much the sooner. When the sheet becomes overheated it will bulge under the pressure, and this bulging tends to open the hole through which the stay passes and permit the stay end to be drawn through the sheet under a pressure much below its proper holding value. The nuts are made one diameter of stay in length, and the locking-nuts about $\frac{3}{4}$ as long. The screw-threads should be a fine standard V-shaped gauge with rounded corners. As stays are apt to corrode rapidly and are difficult to inspect, they should be proportioned amply heavy.

The load that is carried by a stay depends on the area supported and the pressure. It is difficult to estimate the amount of stiffness due to the flanged edges of sheets, but it is safe to say that they will be self-supporting for a distance of at least 3 inches, and in heavy sheets for considerably more. Furthermore, the tubes will sustain the pressure on the sheets for at least 2 inches beyond

their outer surface, and more than that when the sheets are thick. The net area, then, between the limits defined will be the area that must be sustained by the stays.

For experiments on the holding power of stays, reference is made to "Experiments in Boiler Bracing," by F. J. Cole, Trans. Am. Soc. M. E., Vol. XVIII, 1897.

Rules for Stays.

1. U. S. Board of Supervising Inspectors of Steam-vessels.

The maximum stress in pounds allowable per square inch of cross-sectional area for stays used in the construction of marine boilers, when same are accurately fitted and properly secured, shall be ascertained by the following formula :

$$P = \frac{A \times C}{a}$$

where P = working pressure in pounds ;

A = least cross-sectional area of stay in inches ;

a = area of surface supported by one stay, in inches ;

C = a constant, 6,000, 7,000, 8,000, 9,000, as the case may be ;

C = 9,000 for tested steel stays exceeding 2½ inches in diameter ;

C = 8,000 for tested steel stays 1½ inches and not exceeding 2½ inches in diameter, when such stays are not forged or welded. The ends, however, may be upset to a sufficient diameter to allow for the depth of the thread. The diameter shall be taken at the bottom of the thread, provided it is the least diameter of the stay. All such stays after being upset shall be thoroughly annealed.

C = 8,000 for a tested Huston or similar type of brace, the cross-sectional area of which exceeds 5 square inches ;

C = 7,000 for such tested braces when the cross-sectional area is not less than 1.227 and not more than 5 square inches, provided such braces are prepared at one heat from a solid piece of plate without welds ;

C = 6,000 for all stays not otherwise provided for.

The diameter of a screw stay shall be taken at the bottom of the thread, provided it is the least diameter of the stay.

For all stays the least sectional area shall be taken in calculating the stress allowable.

All screw stay-bolts shall be drilled at the ends with a one-eighth inch hole to at least a depth of one-half inch beyond the inside surface of the sheet. Stays through laps or butt straps may be drilled with larger hole to a depth so that the inner end of said larger hole shall not be nearer than the thickness of the boiler plates from the inner surface of the boiler.

Such screw stay-bolts, with or without sockets, may be used in the construction of marine boilers where fresh water is used for generating steam: *Provided, however,* That screw stay-bolts of a greater length than 24 inches

will not be allowed in any instance, unless the ends of said bolts are fitted with nuts. Water used from a surface condenser shall be deemed fresh water.

Holes for screwed stays must be tapped fair and true, and full thread.

The ends of stays which are upset to include the depth of thread shall be thoroughly annealed after being upset.

The sectional area of pins to resist double shear and bending, accurately fitted and secured in crow-feet, sling, and similar stays, shall be at least equal to required sectional area of the brace. Breadth across each side and depth to crown of eye shall be not less than .85 to .55 of diameter of pin. In order to compensate for inaccurate distribution the forks should be proportioned to support two-thirds of the load, thickness of forks to be not less than .66 to .75 of the diameter of pins.

The combined sectional area of rivets used in securing tee irons and crow-feet to shell, said rivets being in tension, shall be not less than the required sectional area of brace. To insure a well-proportioned rivet point, the total length of shank shall closely approximate the grip plus 1.5 times the diameter of the shank. All rivet-holes shall be drilled. Distance from center of rivet-hole to edge of tee irons, crow-feet, and similar fastenings shall be so proportioned that the net sectional areas through sides at rivet-holes shall equal the required rivet section. Rivet-holes shall be slightly countersunk in order to form a fillet at point and head.

2. *Lloyd's Rule.*

The strength of stays supporting flat surfaces is to be calculated from the smallest part of the stay or fastening; the strain upon them is not to exceed the following limits:

Iron Stays. For stays not exceeding $1\frac{1}{4}$ inches effective diameter, and for all stays which are welded, 8000 pounds per square inch. For unwelded stays above $1\frac{1}{4}$ inches effective diameter, 7500 pounds per square inch.

Steel Stays. For screw stays not exceeding $1\frac{1}{4}$ inches effective diameter, 8000 pounds per square inch; for screw stays above $1\frac{1}{4}$ inches effective diameter, 9000 pounds per square inch. For other stays not exceeding $1\frac{1}{4}$ inches, 9000 pounds per square inch, and for stays exceeding $1\frac{1}{4}$ inches, 10,000 pounds. No steel stays are to be welded.

Stay-tubes. The stress is not to exceed 7500 pounds per square inch.

Note.—The size of angles riveted to flat surfaces to act as stays can be determined by rule for "Flat Surfaces." The stress on any stay is determined by the area supported and the pressure. No allowance is made for any additional strength in the flat surface. If the stays are diagonal to the flat surface supported, the stress in the stay is found by dividing the pressure on total surface supported by the cosine of the angle. The area of small gussets should be made heavier than that required by actual calculation, lest all the stress should come on one edge.

Girders. The rule of the U. S. Inspectors is the same as that of the Board of Trade, except $c=825$ when two or three supporting bolts are fitted, and $c=935$ when four bolts are fitted. The bolts are proportioned by the rules for stays. It is usual to make the thickness $\frac{1}{2}$ the depth in short girders and $\frac{1}{4}$ in long ones.

The rule of the Board of Trade (British) is as follows:

When the tops of combustion-boxes or other parts of a boiler are supported by solid girders of rectangular section, the following formula should be used for finding the working pressure to be allowed for the girders, assuming that they are not subjected to a greater temperature than the ordinary heat of steam, and in the case of combustion-chambers that the ends are properly bedded to the edges of the tube-plate, and the back plate of the combustion-box:

$$\text{Working pressure} = p = \frac{C \times d^2 \times T}{(W - P) D \times L}$$

where W denotes width of combustion-box in inches;

P " pitch of supporting bolts in inches;

D " distance between the girders from centre to centre in inches;

L " length of girder in feet;

d " depth of girder in inches;

T " thickness of girder in inches;

N " number of supporting bolts;

C " $\frac{N \times 1000}{N + 1}$ when number of bolts is odd;

C " $\frac{(N + 1) \times 1000}{N + 2}$ when number of bolts is even.

The working pressure for the supporting bolts and for the plate between them should be determined by the rules for ordinary stays and plates.

Combustion-chamber is the name given to that part behind the bridge-wall in which the gases are expected to mix and burn. The term is, however, applied to different parts according to the design. In general the combustion-chamber should be as large as possible, and many boilers are now being arranged so as to keep the furnace proper away from the boiler, that the combustion may be completed before the products are cooled by the water surfaces.

In the ordinary vertical boiler there is no combustion-chamber other than the furnace. Messrs. Dean and Main have designed a vertical boiler of large size (See *Engineering News*, June 23, 1898) in which the height from grate to tube-sheet is 8 feet.

Good results have been obtained by standing the vertical boiler of ordinary design on top of a brick furnace, in order to procure additional height in the combustion-chamber.

In externally fired cylindrical boilers the combustion-chamber is the space behind the bridge-wall, and as generally set the space is large enough. Where such boilers have return tubes, the distance between the back head and the rear wall of setting is about 18 inches for small boilers and 24 inches to 30 inches for large ones.

In internally fired boilers of the Cornish and Lancashire types the combustion-chamber is the space in the flues behind the bridge, and is as large as can be obtained. In vertical and locomotive boilers the furnace forms the combustion-chamber. Sometimes a fire-brick arch supported on water-circulating tubes is placed over part of the fire, and its action is similar to that of a bridge to mix the gases. The space above might then be called a combustion-chamber.

In Scotch and in Flue and Return-Tube, or sometimes called "Marine" boilers, the combustion-chamber is a box constructed inside the boiler and acts as a connection between the tubes and flues. It is, therefore, called a "back connection." In boilers of the "Marine" type there is often a "front connection" to connect the tubes to the uptake or the liner of the steam-drum, or steam-chimney.

The capacity of the combustion-chamber in Scotch and similar boilers should be not less than that of the furnace-flues entering into it, when the boiler is single-ended, but may be slightly less when double-ended. The back and tube plates are flanged in, and the sides and top are riveted on the outside. This arrangement always leaves the calking edge exposed. The water-space between the sides and bottom of the chamber and the shell should not be less than 3 inches in the clear, but $4\frac{1}{2}$ or 5 inches is much better. In general, the wider the space the better the circulation. Sometimes the bottom of the chamber has to be rounded up between the flues so that a manhole may be provided in the back head. The chamber can then be stiffened by angles running across

the boiler, as ordinary stays cannot be used. The back plate of the chamber is made parallel to the back head in many boilers, but it is better to make it slant slightly that the water-space may be wider at the top to facilitate the separation of steam. The space at the bottom is made about 4 to 6 inches and widens to 8 or 9 inches at the top. The bottom plate should be at least $\frac{1}{8}$ -inch thicker than the sides in order to provide for corrosion. The top plate should be the same as the bottom. The tube-plate is usually about $\frac{3}{8}$ - or $\frac{1}{2}$ -inch, but varies from $\frac{1}{2}$ - to $\frac{3}{4}$ -inch. The back plate is generally $\frac{1}{2}$ - or $\frac{3}{8}$ -inch, but the thickness is dependent on the distance between stay centres, and the stays should be so arranged as to give a thickness as above. If very high pressures are used the thicknesses are slightly increased over the figures just given.

In Scotch boilers with natural draft, one chamber may be used common to all furnaces, but with forced drafts it is better to have separate combustion-chambers. Separate chambers add weight and are more expensive, but facilitate the generation of steam and interfere least with the draft.

In single-ended boilers many engineers place a manhole into the chamber through the rear plate, but this is not necessary if the flues are large enough to admit a man over the bridge-wall. When so made, the rear head and back plate of chamber are flanged, and provision has to be made to drive the rivets. These back entrances are of little value, and unless absolutely necessary had better be omitted, as the joint often causes trouble if not extremely well made.

Riveting. The riveted joints are usually the weakest part in a boiler, and as they often leak and give trouble, the greatest care should be bestowed upon them.

The average riveter requires a space of 24 inches to drive a rivet by hand, but some few experts can close a rivet in 16 inches. Allowance for riveting often determines the spaces that must be made in the design.

In order to be tight, rivets should be driven from the water, steam or pressure side.

The joints may be single-, double-, or treble-riveted, etc., according to the number of rows of rivets driven in each plate. In order to make the joint, the edges of the plates may be lapped, or lapped

and strapped, or butt-jointed with single or double straps, welt-strips or cover-plates (Figs. 73, 74, 75, 76 and 77).

When in multiple rows the rivets may be arranged as "chain" riveting, that is, one directly behind the other, or as "zigzag," that is, staggered. The latter is much the better for boiler work. The

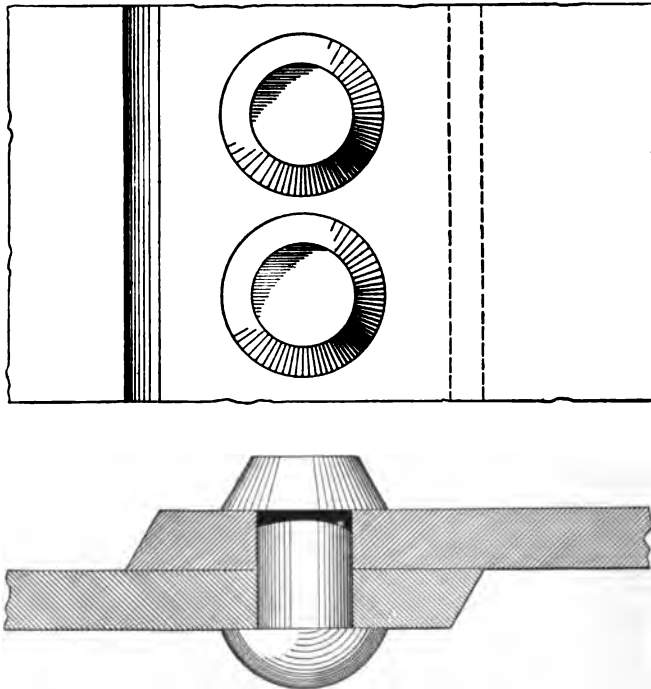


FIG. 73.—Single Riveted Lapped Joint.

"pitch" is the distance between centres of rivets in one row, and the "spacing" is the distance between centre lines of rows.

The "tail" of the rivet is formed on the rivet when purchased, and the "head" is made on the shank end when in place. The shape of the head is either conical or semi-spherical, the latter called a "cup" head. The conical head (Fig. 79) is most common in hand-work. The height of the cone should be $\frac{2}{3}$ or $\frac{3}{4}$ the diameter of shank, and the greatest diameter of the cone about $1\frac{1}{4}$ times the diameter of the shank. If the cone head be not made concentric with the shank, the head will not properly cover the hole in the plate

and will make a weak rivet. As it may be difficult to note the amount of eccentricity, the cone makes a poor form for the head.

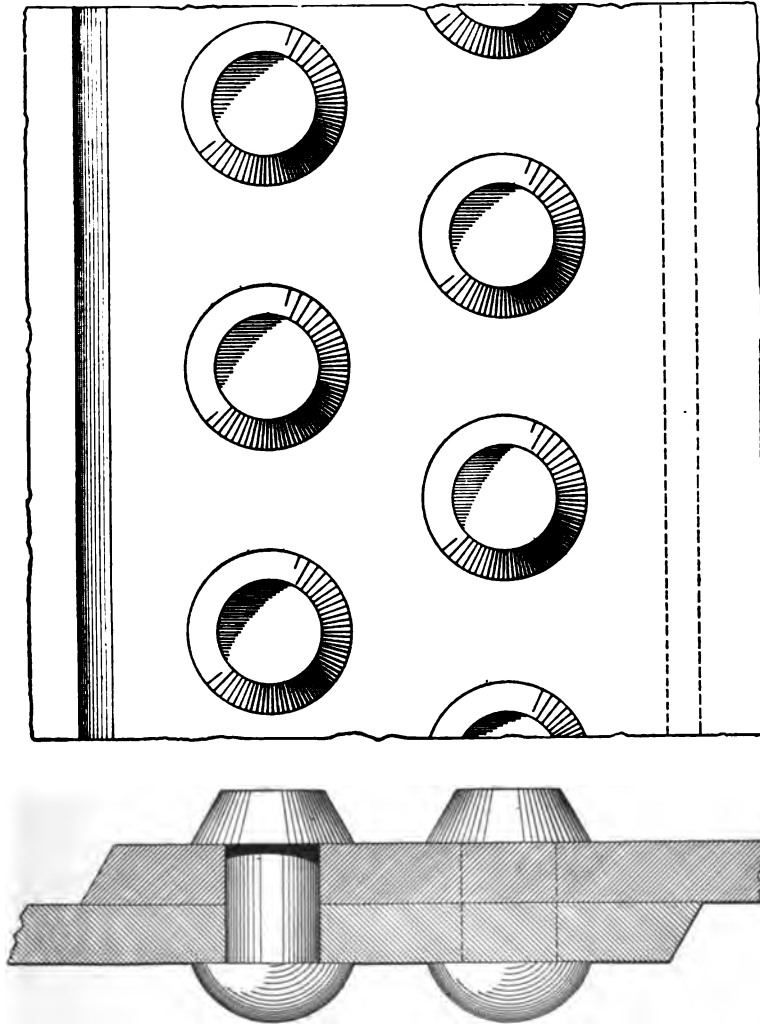


FIG. 74.—Double Riveted Lapped Joint.

It is also a poor shape, due to the thinness of the edges, where it is liable to be rapidly corroded. The spherical or cup head (Fig. 77) is the better form, and is made by a cup-shaped die placed over the

head while being formed. The height of the cup should be $\frac{1}{4}$ or $\frac{1}{2}$ the diameter of shank, and the greatest diameter $1\frac{1}{4}$ times the diam-

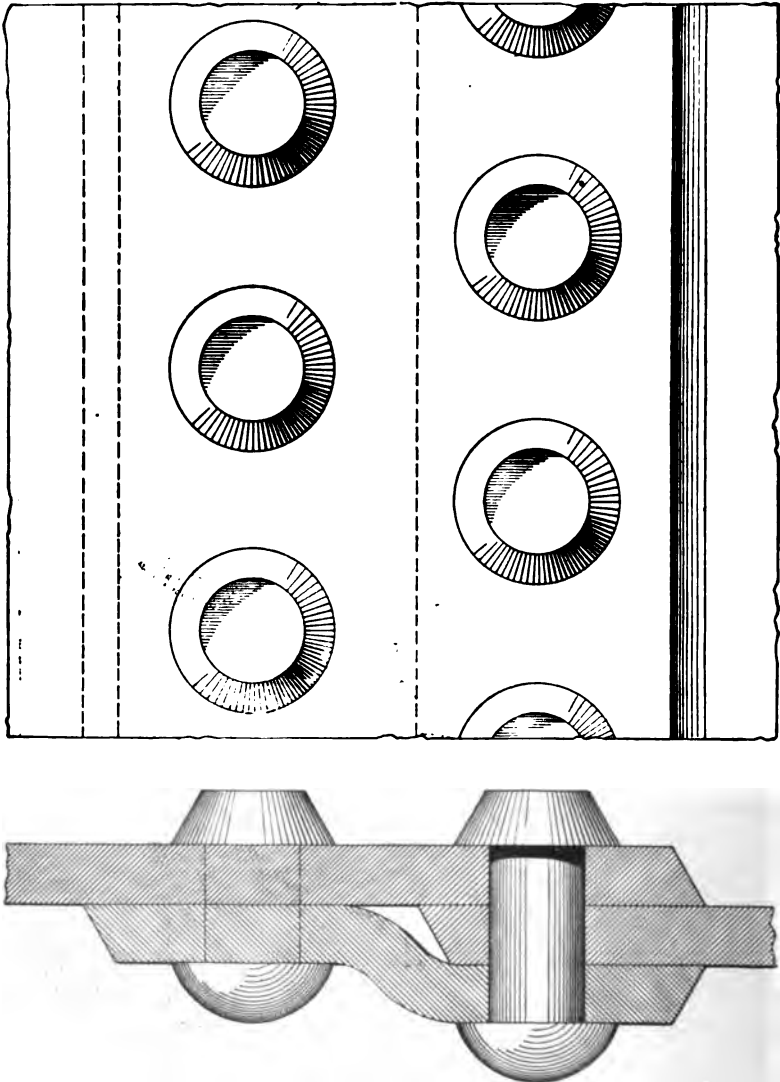


FIG. 75.—Single Riveted Lapped and Strapped Joint.

eter of shank. The bottom edge should not be at right angles to the plate, that is, the head is somewhat less than a semi-sphere. This

permits the edge to be calked more easily, and allows for any surplus metal to flow out from under the die.

The tails are either cup- or button-shaped, the latter sometimes called pan-shaped (Fig. 76).

The allowance of length of shank for forming the head is about $1\frac{1}{4}$ times the diameter for both conical and spherical heads when hand-driven; but about $\frac{1}{8}$ - to $\frac{1}{4}$ -inch more when machine-driven. In addition to the above, there should be an allowance of $\frac{1}{8}$ -inch for each plate when more than two are connected.

Counter-sunk rivets should be avoided as much as possible. The plate is weakened by having so much metal cut away, and the head is more apt to pull through the plate. Counter-sunk heads are liable to leak and are difficult to calk. Such rivets are only permissible when they are placed under flanges or

fittings or in laps not subject to tension, and in direct line with the play of the flames, as, for instance, in the flange of furnace flues (Fig. 78).

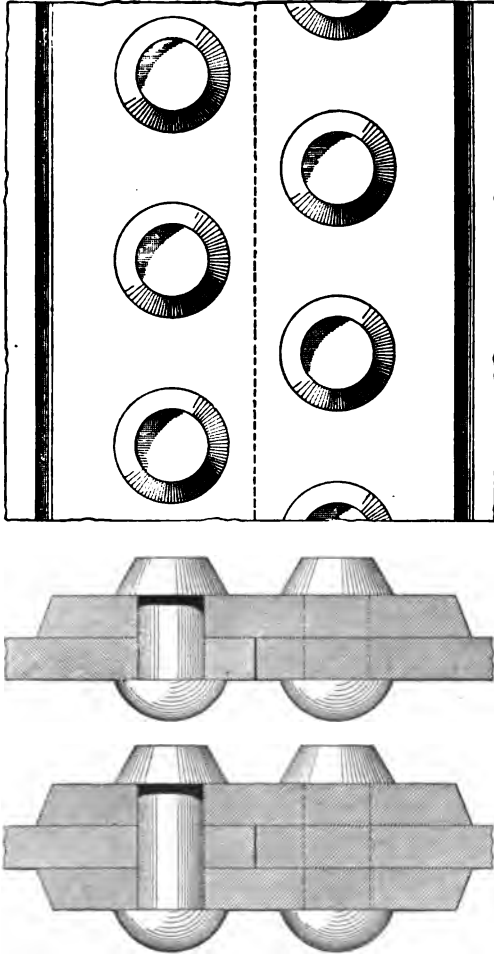


FIG. 76.—Single Riveted Butt Joint, with Single or Double Straps.

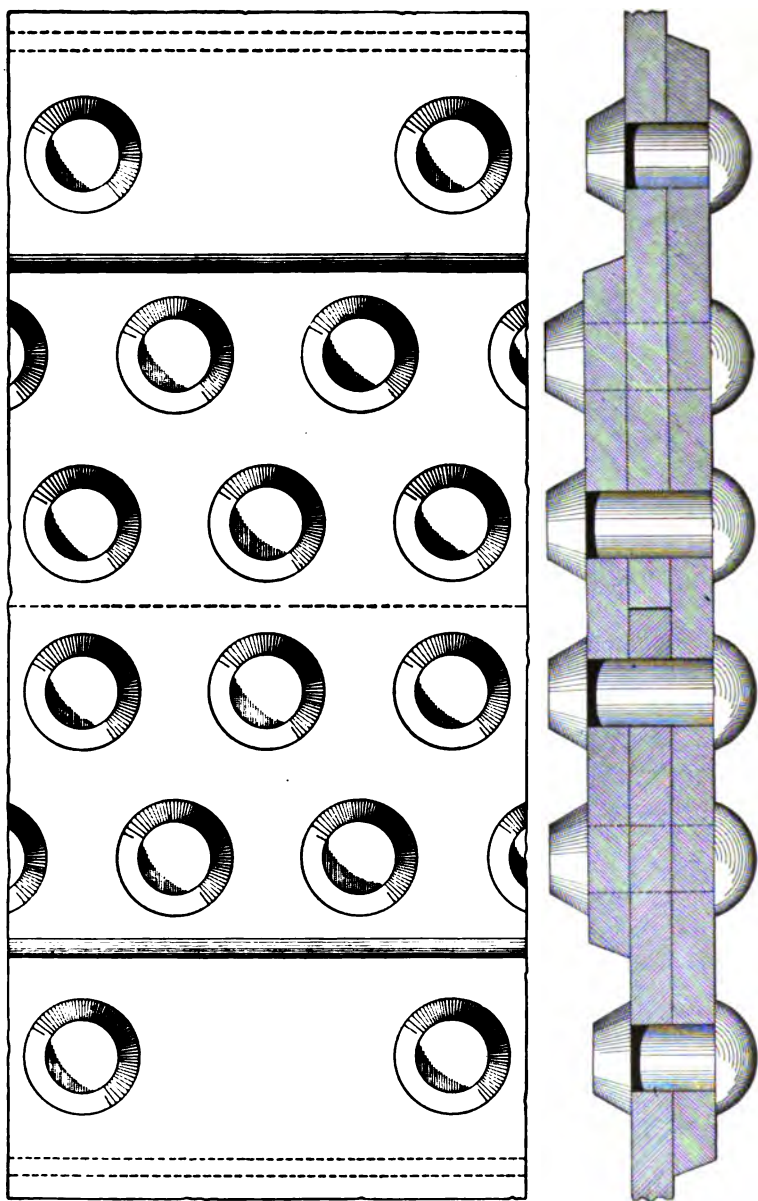


FIG. 77.—Treble Riveted Butt Joint, with straps of unequal widths.

It must be remembered that the object of boiler-riveting is not only to join the plates, but to help form a water-tight joint, and the rivets have to be pitched with this object also in view. The pitch of the outside row of rivets next to the edge of any plate should never exceed seven thicknesses of plate, in order to keep the plates from springing apart between the rivets.* To complete the tightness of the joint, the edges of the plates are calked both inside and

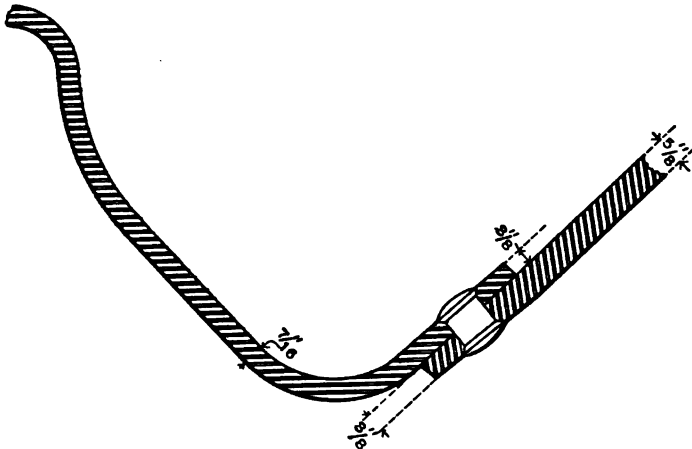


FIG. 78.—Joint between Tube Sheet and Furnace Flue, showing countersunk rivet where exposed to the fire.

outside. The calking tool always should be round-pointed and never square or sharp, as the latter is liable to injure the plates by forming a slight groove, which will increase rapidly by corrosion and by the working of the plates under expansion and contraction. That the calking may be more properly done, the edges of the plates should be planed to a slight bevel.

Rivets always should be driven hot, as they are less liable to be injured than when cold-driven.

Rivets may be driven by hand, but the work is much better performed by power, since the shank is made to more completely fill the hole before the head is formed, and there is less chance of the rivet getting cold due to the quickness of the operation. Against machine-riveting it is sometimes urged that the shank may bulge

* If the joint is at least double-riveted, and the plate $\frac{1}{2}$ -inch or thicker, the maximum pitch may be seven and a half thicknesses.

under the heavy pressure and force the plates apart. While this objection does not seem to be sustained by practice, many of the riveting-machines have a device for holding the plates together during the operation. Machines will satisfactorily close rivets that are so large as to be dangerous to work by hand for fear of creating hidden defects.

Power riveting-machines are operated by either gearing, water, steam or compressed air. In general, the hydraulic machines are preferred, since the work is done more gradually and with less of a shock or blow. The slower and steadier pressure of the water causes the shank to swell throughout its length, thus filling the hole, while under a violent blow the head may be formed first, thus leaving the shank loose. Dr. Coleman Sellers introduced an improvement in the steam-riveter, which is also applicable to compressed air, by adopting a small supply-pipe. This prevents the piston from advancing rapidly and causes it to act in imitation of the hydraulic ram. The hydraulic pressure usually maintained is 1500 to 1600 pounds per square inch, or about 100 atmospheres.

The pressure put upon a rivet should be in proportion to the size of rivet, and machines are made of varying total capacities from a few tons to 150 tons, which is about the heaviest machine in use at the present time. If too great a pressure be placed upon the rivets, the plates will stretch along the seam. On the other hand, in order to insure tight work, boiler-rivets require a heavier pressure than ordinary rivets as used in structural work. For hot rivets, a pressure of 50 tons or 100,000 pounds per square inch of area has been found ample when the rivets are short, and a slightly greater pressure with a slower movement of the ram when the rivets are long. For $\frac{3}{4}$ -inch rivets driven cold, a pressure of 15 tons or 30,000 pounds has been found sufficient, while at 20 tons the metal in the plates has stretched. A pressure of 40 tons on an inch rivet driven hot has given good results, although many use a lower pressure. A machine capable of exerting 60 tons on the rivet is generally considered amply heavy for the largest-sized rivet likely to be used in ordinary boiler construction.

The strength of rivets to resist shearing is sometimes erroneously taken as equal to the tensile strength of the metal. The shearing strength of iron rivets may be taken as 80 per cent of the tensile strength, and of steel rivets as 75 per cent. Rivets in double shear

are from 1.75 to 1.80 times as strong as those in single shear. Resistance to failure by shearing is increased by the friction between the plates. This friction may amount to three or even seven tons per rivet. No allowance is made for this friction in determining the

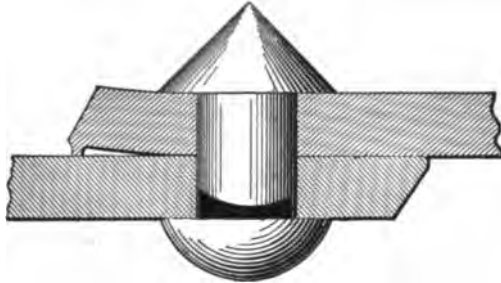


FIG. 79.—Good and Bad Calking.

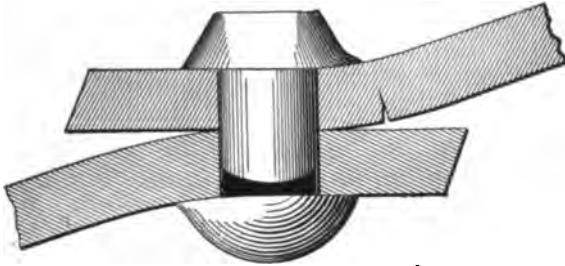


FIG. 80.—Effect of Indirect Pull on a Lapped Joint.

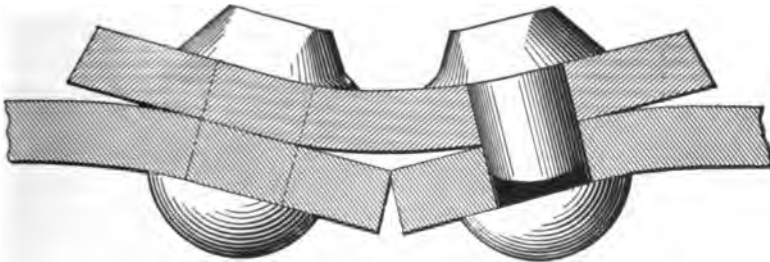


FIG. 81.—Effect of Indirect Pull on a Single Strapped Joint.

strength of joint, as it is very unreliable. In all lap-joints the tendency, due to the indirect pull, is to open the joint and neutralize the friction. Imperfect calking is apt, especially with thin plates, to open the seam (Fig. 79). The effect of this indirect pull on lapped joints is shown in Figs. 80 and 81. With thick plates the tendency to distort the joint will be greater. In double-

strapped butt-joints the force acts in a straight line, and there is no tendency for the joint to open. All longitudinal seams in important boilers, therefore, should be double-strapped.

The continual bending action in lapped joints and in single-strapped butt-joints often causes cracks to form beneath the lap,* as shown in Fig. 82. These cracks widen by corrosion, and being

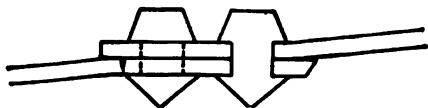
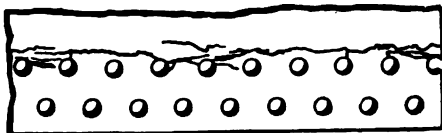
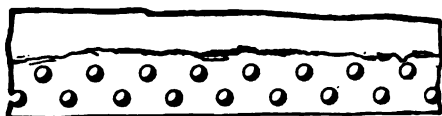


FIG. 82.—Cracks in Lapped Joint due to Bending.

always hidden under the lap, are very dangerous. When single straps are used, they always should be applied on the outside of the shell. If placed internally, the tendency to open at the butt is increased.

Long rivets in boiler work should be avoided, as the heads are liable to fly off after the rivets have been driven, probably due to internal stress while cooling. The greatest mass of metal

is where the head joins the shank, and this part retains the heat longest. Then as the rivet cools and contracts, stresses are set up at this point which may at any time cause the failure of the rivet. These stresses are increased by the length of shank. Rivets never should be driven through more than three full thicknesses of plate, and then only with the greatest care.

Boiler-rivets should be tested for tightness by striking the head a sharp blow with a hammer while the thumb is placed on rivet and forefinger on plate. Any slight movement can then be detected after a little experience.

The rivet-hole is made $\frac{1}{8}$ inch larger than the cold rivet, so that it may be inserted when hot.

It is of great importance that the holes should match fair and not overlap or be "partly blind." If they be unfair, the rivets will not fill the holes and a leak is apt to occur. The holes should

* Many sample cases are illustrated in *Locomotive* (Hartford Steam Boiler Inspection and Insurance Co.), January, 1897.

never be forced to match by using a drift-pin, which only weakens the plates by stretching them beyond their elastic limit. When the holes are not fair it is best to ream them out and use a larger rivet.

The holes are either punched or drilled. Wrought iron and the mildest grades of steel are not seriously affected by punching. Ordinary boiler steel, especially the harder grades, is injured by punching, and the effect extends a variable distance, according to the thickness of plate and size of hole. As the injury is not visible and can only be made apparent by etching with acid, a test not practicable in boiler construction, even the mildest steel should not be punched lest an undetected flaw be created. Steel plates should be drilled and have the burr formed by the drill removed. There is little danger, however, in punching thin, mild steel plates and then reaming, so as to remove at least $\frac{1}{16}$ -inch of metal around the hole. The best practice is to drill both plates together after they have been rolled to shape, and thus insure perfect fairness of holes. Drilling is more accurate than punching, but is slower, while punching is cheaper and practised by some boilermakers on that account.

From tests made on steel plates, it appears that thin plates suffer least from punching, but when thicker than $\frac{3}{8}$ -inch, the loss due to punching varies from 6 per cent to 33 per cent. This loss may be partially, and in some cases totally, removed by subsequent annealing. It is claimed that rivets are more easily sheared in drilled than in punched holes, but this is probably not the case if the sharp corners of the drilled holes be slightly rounded with a file, as they should be in all important work.

The end of the punch should be slightly concave, with the diameter of the cutting edge a trifle larger than the shank, so as to make a clean cut. The hole in the die is somewhat larger than the punch, in order that no resistance may be offered to the action of the punch



FIG. 83.—Rivets in Punched Holes.

or the discharge of the "wad." The diameters are usually in the ratio of 1 to 1.1 or 1.2. This produces a conical-shaped hole. The plates should be put together so that the small diameters are at the centre. The swelling of the shank of the rivet will then assist the head in holding the plates. If placed otherwise the swelling of the rivet may wedge the plates apart (Fig. 83).

The strength of a riveted joint is always less than that of the plate, on account of the plate being weakened by the metal cut away by the holes. The strength of the joint should be calculated and a proper factor of safety used to determine the allowed safe pressure. A long line of rivets is considered stronger than a few as used for testing purposes, and also the plate between the holes is considered stronger than its actual tensile strength. These elements of strength are not considered in the calculation, as they are too variable and difficult to credit with proper values.

A riveted seam fails in one of the following ways:

1. By the shearing of the rivets;
2. By the plate breaking between the holes;
3. By the plate breaking between the holes and edge;
4. By the plate crushing in front of the rivet;*
5. By the plate shearing in front of the rivet.

In general a line of fracture includes more than one of the above five conditions, for when the rupture once takes place it often does not have time to follow the line of least resistance and new stresses are brought to bear, so that it frequently becomes difficult from an examination of the rent to determine just where the break first occurred.

In order to supply ample strength to resist the third, fourth and fifth conditions, practice dictates that the distance from hole to edge of plate should be at least equal to diameter of rivet.

Plates usually fail first by tearing between the rivet-holes, caused by brittleness of plate, injury due to punching, expansion and contraction stresses and loss of section due to corrosion. As the rivet is protected under ordinary conditions from corrosive effects, it is well to design new boilers so that there shall be a small increase in the strength of plate between holes over the shearing strength of rivets.

The strength of the joint is calculated for every possible class of failure. The lowest strength so found when compared with the strength of solid plate, expressed in percentage, is called the efficiency of the joint.

The method of calculating the efficiency of riveted joints, when

* The crushing strength can be taken at 90,000 lbs. to 95,000 lbs. per square inch on an area equal to the diameter of rivet-hole times the thickness of plate.

the rivets are properly spaced back from the edge and between rows, is as follows:

SINGLE-RIVETED JOINT

Steel plate, tensile strength per square inch of section, 60,000 lbs.;

Thickness of plate, $\frac{3}{8}$ inch = 0.375;

Diameter rivet-holes, $\frac{1}{8}$ in. = 0.8125;

Area of rivet-hole = 0.5185 sq. in.;

Pitch of rivets, $1\frac{1}{4}$ in. = 1.875;

Shearing resistance of steel rivets per square inch = 45,000 lbs.;

Then $1.875 \times 0.375 \times 60,000 = 42,187$ = strength of solid plate;

$(1.875 - 0.8125) \times 0.375 \times 60,000 = 23,906$ lbs. = strength of net section of plate;

$0.5185 \times 45,000 = 23,332$ lbs. = strength of one rivet in single shear.

The rivet strength is the weakest; therefore $23,332 \div 42,187 = 55.3$ per cent efficiency of joint.

DOUBLE-RIVETED JOINT

In double-riveted lap-joints an accession of strength is found over the single-riveted joint of about 20 per cent. This arises from the wider lap and the better distribution of the material. The rivets are pitched wider, and there is more rivet area to be sheared, together with a larger percentage of net section of plate to be broken.

Steel plate, tensile strength per square inch of section, 60,000 lbs.;

Thickness of plate, $\frac{3}{8}$ inch = 0.375;

Diameter of rivet-holes, $\frac{1}{8}$ in. = 0.8125;

Area rivet-hole = 0.5185 sq. in.;

Pitch of rivets = 2.5 in.;

Shearing resistance of steel rivets per square inch, 45,000 lbs.;

Then $2.5 \times 0.375 \times 60,000 = 56,250$ lbs. = strength of solid plate;

$(2.5 - 0.8125) \times 0.375 \times 60,000 = 37,969$ lbs. = strength of net section of plate;

$0.5185 \times 2 \times 45,000 = 46,665$ = strength of two rivets in single shear.

Net section of plate is the weakest; therefore $37,969 \div 56,250 = 67.5$ per cent efficiency of joint.

TRIPLE-RIVETED JOINT

In a triple lap-riveted joint there is a gain in strength for reasons similar to those above.

Steel plate, tensile strength per square inch of section, 60,000 lbs.;

Thickness of plate, $\frac{3}{8}$ in. = 0.375;

Diameter of rivet-holes, $\frac{1}{2}$ inch = 0.8125;

Area one rivet-hole = 0.5185 sq. in.;

Pitch of rivets = $3\frac{1}{2}$ in. = 3.5;

Shearing resistance of steel rivets per square inch, 45,000 lbs.;

Then $3.5 \times 0.375 \times 60,000 = 78,750$ lbs. = strength of solid plate;

$(3.5 - 0.8125) \times 0.375 \times 60,000 = 60,469$ lbs. = strength of net section of plate.

$0.5185 \times 3 \times 45,000 = 69,997$ lbs. = strength of three rivets in single shear.

Net section of plate is the weakest, and efficiency is 76.7%.

DOUBLE-RIVETED, DOUBLE-STRAPPED BUTT-JOINT

This joint is calculated the same as a double-riveted joint, except that the shearing strength of the rivets is increased for double shear.

TRIPLE-RIVETED DOUBLE-STRAPPED BUTT-JOINT

When inner and outer straps have the same width.

Steel plate, tensile strength per square inch of section, 60,000 lbs.;

Thickness of plate, $\frac{3}{8}$ inch = 0.375;

Diameter of rivet-holes = $\frac{1}{2}$ inch = 0.8125;

Area of rivet-holes = 0.5185 sq. in.;

Pitch of rivets, $3\frac{1}{2}$ inches = 3.5;

Resistance of steel rivets in double shear, 78,750 lbs.;

Then $3.5 \times 0.375 \times 60,000 = 78,750$ lbs. = strength of solid plate;

$(3.5 - 0.8125) \times 0.375 \times 60,000 = 60,469$ lbs. = strength of net section of plate;

$0.5185 \times 3 \times 78,750 = 122,495$ lbs. = strength of three rivets in double shear.

Net section of plate is the weakest, and efficiency is 76.7%.

This is not a well-proportioned joint, because the rivet strength is too great. The rivets should have been spaced farther apart, although there is no advantage in treble riveting plates as thin as

$\frac{3}{8}$ -inch. The selection was made simply to show the method employed and to afford comparison with the next case.

When the inner strap is wider than the outer strap, and the pitch of the outside row of rivets is twice that of the inside rows. The rivets in the outside row now are in single shear. This arrangement increases the net section of plate* and reduces the area of rivets to be sheared, thereby increasing the efficiency of the joint.

Steel plate, tensile strength per square inch of section, 60,000 lbs.;

Thickness of plate, $\frac{3}{8}$ inch = 0.375;

Diameter of rivet-holes, $\frac{1}{4}$ inch = 0.8125;

Area of rivet-hole = 0.5185 sq. in.;

Pitch of rivets in inner rows, $3\frac{1}{2}$ inches = 3.5;

Pitch of rivets in outer rows, 7 inches;

Resistance of steel rivets in single shear, 45,000 lbs.;

Resistance of steel rivets in double shear, 78,750 lbs.;

Then $7 \times 0.375 \times 60,000 = 157,500$ = strength of solid plate;

$(7 - 0.8125) \times 0.375 \times 60,000 = 139,219$ = strength of net section of plate;

$0.5185 \times 4 \times 78,750 = 163,327$ = strength of four rivets in double shear;

$0.5185 \times 45,000 = 23,333$ = strength of one rivet in single shear;

$163,327 + 23,333 = 186,660$ = shearing strength of all five rivets.

Net section of plate is the weakest, and efficiency is 88.4%.

This style of joint is much used on the longitudinal seams of large boilers made of heavy plates.

Many engineers prefer to modify the above methods by making an allowance for the increase in strength of plate between perforations over that of the plain plate. This increase is similar to that found in short test specimens over long ones. It varies from perhaps 5 per cent to over 20 per cent, and depends on the length between holes and on the thickness. How much of this increase can be fairly trusted in a long seam is doubtful, due to possible unfairness in matching holes, and to the unequal loads on the rivets.

Many boilermakers determine the pitch by trial, using the following formula:

* Because the net strength of plate is between the outer row of rivets. If not, the outer row has to shear with the breaking of plate between inner row.

(a). $\frac{(p-d) \times 100}{p}$ = percentage of strength of plate at joint, as compared to solid plate;

(b). $\frac{A \times N \times C \times 100}{p \times t}$ = percentage of strength of rivet, as compared to solid plate;

in which A denotes area of one rivet-hole in square inches;

d " diameter of rivet-holes in inches;

p " pitch or distance between centres of holes in inches;

t " thickness of plate in inches;

N " number of rivets sheared;

C " 1.00 for single shear and 1.75 for double shear.

Since the shearing strength of rivet cannot be taken as equal to the tensile strength of plate, and also since the holes are not always drilled fair and true, it is found best to so design the joint that the percentage of rivet strength should exceed that of plate at joint in about the following proportions:

For iron rivets, as 12 to 8;

For steel rivets, as 28 to 23.

In other words, the result of equation (b) to that of equation (a) should be in about the above proportion.

The size of rivet should depend upon the thickness of plate, although practice has become more or less uniform in the use of certain sizes for different plates. No doubt higher efficiencies would be obtained by using larger rivets in the thicker plates than are commonly adopted. Boiler-rivets are seldom used of larger size than $1\frac{1}{4}$ inches in diameter, owing to the difficulty of driving them. American practice rarely uses rivets between the even $\frac{1}{2}$ inch in diameter, although foreign builders adopt the intermediate sizes varying by $\frac{1}{16}$ inch, a practice which has much to commend it.

It is always most convenient for manufacturing reasons to design, whenever possible, all joints in the boiler with the same-sized rivets.

Table XVIII represents about the average practice of boiler-shops, showing the size of rivet and pitch. When iron rivets are used, the pitch can be reduced. The figures have been adopted for simplicity and uniformity, rather than for producing the strongest possible combination. The low efficiency in some cases is probably

more than offset by the decrease in risk of ruining a plate by incorrect drilling, which might occur in a shop if no standard were adopted.

TABLE XVIII
DETAILS OF RIVETED JOINTS

Thick- ness of Plate.	Diam- eter of Rivet.	Single-riveted Lap-Joint.		Spac- ing.	Double-riveted			
		Pitch.	Efficiency.		Lap-joint.		Double-strap Butt.	
					Pitch.	Efficiency.	Pitch.	Efficiency.
$\frac{1}{8}$	$\frac{1}{8}$	1 $\frac{1}{2}$	54.0%	1	1 $\frac{1}{2}$	67.9%	2 $\frac{1}{2}$	76.6%
$\frac{1}{4}$	$\frac{1}{4}$	1 $\frac{1}{2}$	54.6%	1 $\frac{1}{2}$	2 $\frac{1}{2}$	67.6%	2 $\frac{1}{2}$	75.3%
$\frac{3}{8}$	$\frac{3}{8}$	1 $\frac{1}{2}$	55.3%	1 $\frac{1}{2}$	2 $\frac{1}{2}$	67.5%	3	73.0%
$\frac{1}{2}$	$\frac{1}{2}$	2	51.7%	1 $\frac{1}{2}$	2 $\frac{1}{2}$	69.0%	3 $\frac{1}{2}$	72.7%
$\frac{5}{8}$	$\frac{5}{8}$	2	51.7%	1 $\frac{1}{2}$	2 $\frac{1}{2}$	67.2%	3 $\frac{1}{2}$	72.2%
$\frac{3}{4}$	$\frac{3}{4}$	2 $\frac{1}{2}$	50.8%	1 $\frac{1}{2}$	3	66.6%	3 $\frac{1}{2}$	73.6%
$\frac{7}{8}$	$\frac{7}{8}$	2 $\frac{1}{2}$	47.1%	1 $\frac{1}{2}$	3	62.8%	3 $\frac{1}{2}$	73.4%
1	1	2 $\frac{1}{2}$	41.6%	1 $\frac{1}{2}$	3	57.2%	3 $\frac{1}{2}$	73.3%
$1\frac{1}{8}$	1	2 $\frac{1}{2}$	37.4%	2	3 $\frac{1}{2}$	57.0%	4	72.8%
$1\frac{1}{4}$	1	2 $\frac{1}{2}$	34.4%	2	3 $\frac{1}{2}$	52.4%	4	71.5%
$1\frac{1}{2}$	1	2 $\frac{1}{2}$	31.8%	2	3 $\frac{1}{2}$	49.0%	4	66.5%
$1\frac{3}{4}$	1 $\frac{1}{4}$	2 $\frac{1}{2}$	32.0%	2 $\frac{1}{2}$	3 $\frac{1}{2}$	47.2%	4 $\frac{1}{2}$	65.6%
1	1 $\frac{1}{2}$	2 $\frac{1}{2}$	30.0%	2 $\frac{1}{2}$	3 $\frac{1}{2}$	47.4%	4 $\frac{1}{2}$	64.6%
$1\frac{1}{4}$	1 $\frac{1}{2}$	2 $\frac{1}{2}$	28.2%	2 $\frac{1}{2}$	3 $\frac{1}{2}$	44.6%	4 $\frac{1}{2}$	60.8%

Note. This table shows the necessity, when using thick plates, of double riveting lap-joints and of treble-riveting double-strap butt-joints, in order to secure high efficiencies.

The efficiencies were calculated by the author on the assumption of steel plate with tensile strength of 60,000 lbs., steel rivets with shearing strength of 45,000 lbs. for single and 78,750 lbs. for double shear, and rivet-holes $\frac{1}{16}$ inch larger than rivet.

The distance from edge of plate to centre of first row of rivets should be one and one-half diameters of rivet.

The width of lap for single-riveting should be three diameters of rivet.

The width of lap for double-riveting should be five diameters of rivet.

The width of single-riveted butt-strap should be six diameters of rivet.

The width of double-riveted butt-strap should be ten diameters of rivet.

A single butt-strap must not be less than the thickness of the plate.

A double butt-strap must not be less than five-eighths the thickness of plate.

When plates of unequal thickness are to be joined, then the thickness of the thinnest plate should be used to determine the variable dimensions, such as diameter of rivet, spacing back from edge, thickness of butt-strap, etc.

It is at times necessary to have a butt-strap on a surface that is supported by stay-bolts, and such cases are more or less difficult to design. The Bigelow Boiler Company, of New Haven, Conn., adopts a form shown in Fig. 84 for the water-legs of some of their

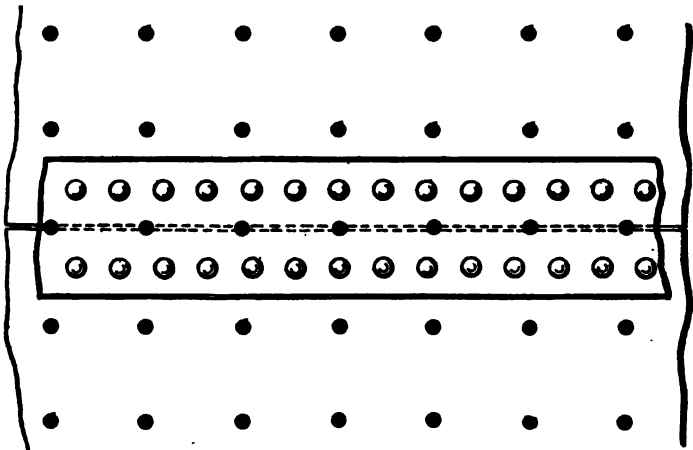


FIG. 84.—Butt-strap on a Stayed Sheet.

upright tubular boilers. A form of triple-riveted butt-joint with unequal straps is shown in Fig. 85, taken from *The Locomotive*, May, 1898. The stay-bolts are shown in black. From an experiment made by the Bigelow Company and the Hartford Steam Boiler and Insurance Company, described in the issue of *The Locomotive* mentioned, the conclusion was drawn that it was not necessary, except in special cases, to provide a triple-riveted butt-joint on the outer sheet of a curved and properly stayed water-leg.

Fig. 86 illustrates a design for a triple-riveted butt-strap.

Welding. Boiler-sheets have been joined by welding. Although only used to a limited extent, this method has many promising advantages. The principal objection is the cost.

The strength of the weld, when well done, appears to be equal

to that of the sheet, but always cannot be relied upon. For making domes, cylinders for storing gases under heavy pressures, and for special shapes, welding has been very successful.

The welding may be accomplished by the use of an electric current or by heating the edges and pressing them together. This is generally done by passing the part to be welded between rollers. Lest the plates should stretch under the latter opera-

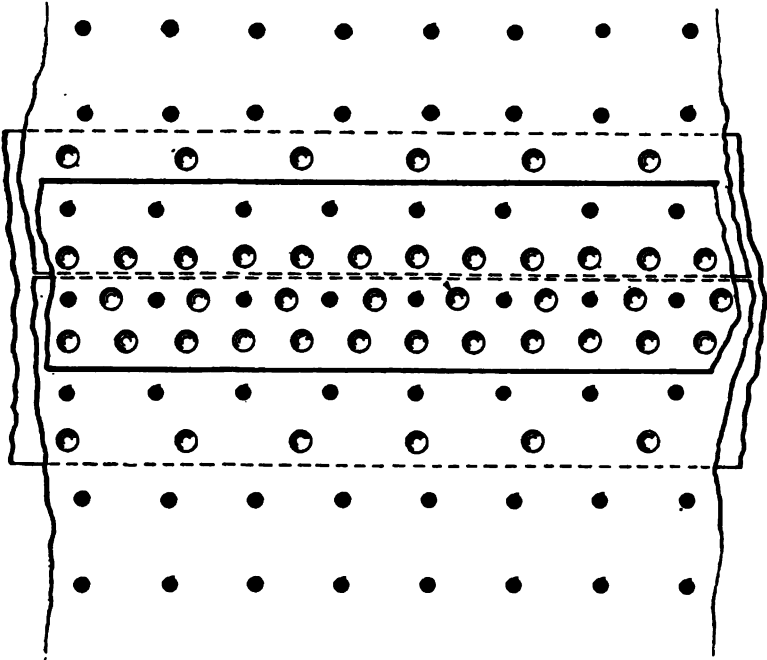


FIG. 85.—Double-riveted Butt-strap of Unequal Width on a Stayed Sheet.

tion, they should be welded first at the centre and then at the ends, the intermediate portion being welded afterward. Corrugated and similar forms of flues and boiler-tubes are always lap-welded. Electric welding is done by the use of an alternating current of low voltage, generally not exceeding three volts, and of large volume.

The edges should be slightly upset or thickened and bevelled, and be heated on both sides at once. The pieces may be heated in a special gas-furnace, using a mixture of $2\frac{1}{2}$ volumes of air to one volume of hydrogen or water gas. The use of a "glut"

piece is discarded as unnecessary and as only extending the surface of the weld.

A general use of welding for boiler shells subject to tension has been prevented by the irregular strengths of the welds, the efficiency varying from about 50 per cent to 100 per cent. Steel cannot be always successfully welded, which is true unless the material be *mild*. The danger of the use of a sheet that will not weld, the

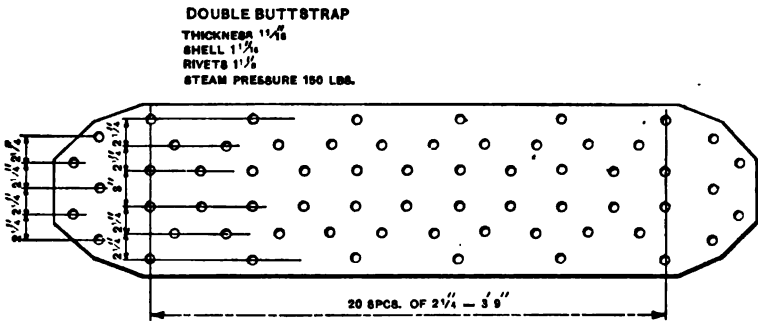


FIG. 86.—Design for a Triple-riveted Butt-strap.

formation of a coating of oxide, and the failure to show visible evidence of defect must always carry great weight against welding as at present practised.

Setting. Boilers of all types should be set on a firm foundation so as to prevent unequal settling. They always should be located in dry places.

Most boilers are set in brickwork, although some of the water tubular types are encased in sheet iron or steel, and a few special designs, as the upright or vertical boilers, require no setting beyond the foundation.

However set, all boilers must be allowed ample freedom for expansion and contraction, lest the setting be seriously damaged. The brickwork is often laid so close to the shell that the rivet-heads are encased. When expansion takes place, some of the bricks are dislodged. On contracting again, dirt settles in the newly made opening, and the process is repeated until the setting is badly cracked.

Good hard-burned bricks should be used, set in strong mortar. The brickwork should not touch the boiler, as the bricks are hygroscopic and retain moisture for a long time, thus rendering the

boiler liable to corrosion at a place not easily seen. A better plan is the filling of all spaces between the boiler and the masonry with asbestos and fire-clay.

The setting of each boiler should be designed for its particular kind and location, but a few general hints may not be out of place. The setting should have more attention given to it than is usually done, and less reliance placed on the mason, who has little interest at stake. Considerable quantities of air filter through even good brickwork, which materially damages the draft and general efficiency. All doors and other openings should have good cast-iron frames, so set in the wall as to be practically airtight between frame and bricks. The doors should fit snug. Since the draft is always inward, the leakage is not readily visible or determinable. It is a good practice to paint the brick setting with some heavy tar paint.

The brick walls are best made double with an air space of 2 inches between them. These walls can be tied across at intervals by headers. A very good plan is to make the outer wall $12\frac{1}{2}$ inches thick and the inner wall 8 inches with an air space of 4 inches. The headers in the outer wall project across the air space and simply touch the inner wall, but are not tied to it. This arrangement permits the two walls to expand at will without injury to each other, while the headers lend support to the inner one.

The joints should be about $\frac{1}{4}$ -inch thick. The mortar is frequently of lime, but should be of hydraulic cement, or one part cement to three parts lime.*

When single walls without an air space are used, they should be two bricks or $17\frac{1}{2}$ inches thick, exclusive of lining. When boilers are set in battery, the partition or division walls need be only $12\frac{1}{2}$ inches in thickness. The outer walls are tied together by tie-rods about one inch in diameter and fastened to binder bars or brick staves. These binder bars are generally made of cast-iron with a tee section, having the greatest depth at the centre. The ends of the walls should be exposed, so that any bulging may be quickly noticed.

The inside of the brickwork exposed to the direct action of the heat should be of fire-brick 4 inches or $4\frac{1}{2}$ inches, according to the size of brick used, and be set in fire-clay. If any trimming has to be done, trim the red bricks in preference to the fire-bricks. This fire-brick lining should be arranged independent of the

* The lime makes the cement work more smoothly and set more slowly.

regular brick setting, so that it may be renewed without necessitating the taking down of the latter.

The tops of many externally fired boilers are covered with a brick arch resting on the side walls. This is not a good plan, as leaks may occur and not be noticed. A better plan is to cover the top with sectional covering blocks laid touching the boiler, so that if a leak occurs a wet spot will show; or with sand which can be brushed aside for inspection. To prevent the sand running out, the joint between the shell and the setting, which is liable to open by expansion, should be filled with asbestos and be covered with sheets of asbestos paper well lapped.

Scotch boilers do not require a brickwork setting. They rest on saddles placed on suitable foundations. In ships these saddles are either of steel plates or of cast-iron, or are formed by extending upward the ship's floors and riveting double angles on the tops, curved to fit the shell. In stationary work the saddles are nearly always of cast-iron. In length they should be not less than one-third the diameter of shell and about 5 to 7 inches in width. Ordinarily two saddles are sufficient; and three should only be used with great care, owing to the difficulty of keeping these points in alignment, so as to divide the load. When the saddles are large they may be made in halves and bolted together. When boilers are rested on metallic supports it is customary to place red lead or putty on the supports, in order that an even surface may be insured. It should be used thick, so that the boiler may squeeze it down to a proper bearing. Pure white-lead putty, well mixed with a good oil, is much better than red lead, as the latter gets hard and brittle and chips out. The boiler need not be fastened to the saddles, as its weight is sufficient. On shipboard the boiler must be tied **down both** vertically and fore and aft, to prevent dislodgement due to rolling, pitching or collision. The size of these steel tie-pieces cannot be calculated, but are made to suit the conditions and judgment of the designer.

It is often convenient to estimate the approximate number of bricks required for the setting of a horizontal return-tubular boiler, and the accompanying table, taken from *Locomotive*, November, 1891, will be found useful.

Bridge Wall. At the back end of the grate a bridge wall is formed so as to prevent the coal from falling off, and to compel the draft to pass upward through the grate and bed of coal.

TABLE XIX
NUMBER OF BRICKS IN BOILER SETTINGS

Diameter of Boiler in Inches.	Length of Tubes in Feet.	Kind of Front.	Number of Common Brick.	Number of Fire-brick.	Additional Number of Brick per Foot in Length.	Number of Common Brick for Each Boiler After the First.	Additional Number of Brick for Each Boiler After the First per Lin. Foot of Tubes.
36	10	Flush	11,700	564	660	6,300	350
36	10	Overhanging	11,000	525	660	5,900	350
42	12	Flush	13,700	654	680	7,400	360
42	12	Overhanging	13,000	629	680	7,000	360
48	15	Flush	16,700	850	710	8,900	370
48	15	Overhanging	16,000	816	710	8,500	370
54	15	Flush	17,600	990	730	9,400	380
54	15	Overhanging	16,700	886	730	8,900	380
60	16	Flush	19,100	1,140	760	10,200	400
60	16	Overhanging	18,200	950	760	9,700	400
66	16	Flush	21,900	1,290	830	11,900	450
66	16	Overhanging	20,600	1,080	830	11,300	450
72	18	Flush	24,000	1,400	860	13,400	460
72	18	Overhanging	23,000	1,150	860	12,800	460

In externally fired boilers the wall is built of brick lined on the fire side and top with fire-brick. In internally fired boilers the wall is usually made of two or three pieces of special-shaped fire-brick cemented with fire-clay, or of ordinary fire-brick and fire-clay.

The shape of the wall, whether flat on top or curved to correspond with round of shell, with vertical or with sloping sides, appears to make little difference, according to tests made by George H. Barrus. The area over the wall must be large enough so as not to check the draft, while beyond that the effect of shape appears to be slight. A flat wall is easier to build, but most engineers prefer a curved top and a vertical front face, with the upper edge cut away at an angle of 45 degrees.

With soft and hydrocarbonaceous coals it is best to admit some air above the grate, and for that purpose the bridge wall is often made "split," that is, hollow, with air-passages in its back face or on top. These passages or holes may be made in a cast-iron plate set in the bridge wall, or be made between the bricks by spacing them a short distance apart. The hollow centre of the wall can be connected to the air space in the side walls of setting, so as to

draw heated air only. The air-supply should be easily controlled by a damper. In internally fired flues, air may be passed from the ash-pit through an opening in the plate beneath the bridge wall, which opening can be controlled by a slide or damper door, easily moved by the slice-bar or poker from the front. (Figs. 124 and 125.)

The split bridge often materially assists in preventing the generation of an excess of smoke, but, like every other device, must be intelligently handled.

CHAPTER IX

BOILER FITTINGS

Mountings and Gaskets. Steam-dome. Steam-drum. Steam-superheater. Steam-chimney. Steam-pipe. Stop-valve. Dry Pipe. Boiler-feed. Injector and Pump. Feed-water Heater, Purifier, and Economizer. Filters. Mud-drum. Blow-off. Bottom and Surface Blows. Safety-valve. Fusible Plug. Steam-gauge. Water-gauge. Try-cocks. Water-alarm. Man-hole and Hand-hole. Grates, Stationary and Shaking. Down-draft Grates. Ash-pit. Fire-doors. Breeching. Uptake. Smoke Connection. Draft Regulator. Steam-traps. Separators. Evaporators.

IN placing the mountings of a steam-boiler, care must be taken to insure a tight joint and one that will remain so under the trying stresses of usage. Many boilers have undoubtedly failed, while otherwise amply strong, due to carelessness in this regard.

There is little trouble on flat parts, since the flange of the mounting can be faced, and bolted or riveted direct to the sheet. It is well to place between the flange and the plate either cement or a gasket of fine-brass wire netting set in red-lead putty. The flange must be heavy enough to allow the nuts to be screwed up hard, and these bolts must not be spaced too far apart lest the joint be apt to leak. The bolts should not be spaced farther than seven thicknesses of the plate or flange, whichever is the thinner. The nut should be screwed down on a grummet of cement, or cotton waste, or lamp-wicking mixed with red-lead cement. It is best to face off the base of the nut and turn a shallow groove so as to hold the packing.

On the curved surfaces of the shell or dome the mounting may be placed on a seating riveted to the plate. This seating can be made curved to truly fit the plate, and flat to fit the flange of mounting; and can be calked against the plate both on inside and outside of shell.

Small pipes may be screwed into the shell-plates, but it is well to have a flange in all cases. When a flange is not convenient, a

thickening plate may be riveted to the plate and both tapped for the pipe thread.

Corrugated-copper gaskets will be found very serviceable for making large joints, especially for those which have to be occasionally taken apart. Hard-rubber gaskets are frequently used; and, at times, copper wire or small lead pipe laid in a groove in the flange and pressed tight when the bolts are set up, will prove satisfactory. This groove must be on the pressure side of all bolt-holes.

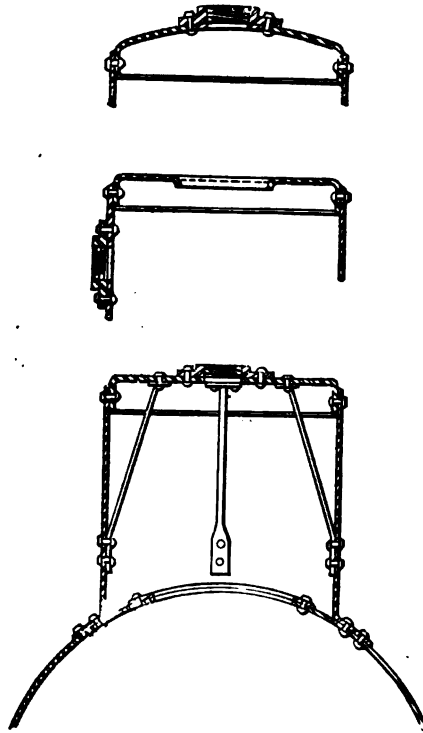


FIG. 87.—Steam-dome.

Steam-domes are common appendages to the ordinary boiler, but are gradually being discarded. They serve to increase the steam space and permit dry steam to collect at a point high above the water-line, whence it may be drawn off by the engine. As a matter of fact, their usefulness for this purpose is rather more

imaginary than real. As usually constructed, they are not sufficiently large to materially affect the steam room, and a few strokes of the engine will exhaust them. Also, judging by the mud and scale that often accumulates within, they are of little aid in furnishing dry steam. A dry pipe or steam-collecting pipe may be used to better advantage.

Domes weaken the shell, due to the large hole that has to be cut out (Fig. 87). The shell should be strengthened at that point, although the strength due to the flanging and fastening of the dome is usually relied upon. When large domes are used, it is only necessary to cut a hole in the shell large enough for a man to pass through, and let the dome attach back from the edge. The edge of the hole should be stiffened the same as if for a manhole. The objection to this plan is the formation of a shelf, caused by the projecting sheet of the shell, on which water and mud will collect. This projection of the shell should be perforated so as to drain, but even so, the drainage is not effectual. If the hole at the centre for steam be too small, there will be a tendency to prime.

The top of the dome can be made out of one sheet, and be flanged to meet the shell of the dome, having the lap on the inside. This top can be bumped so as to be self-supporting, being made as

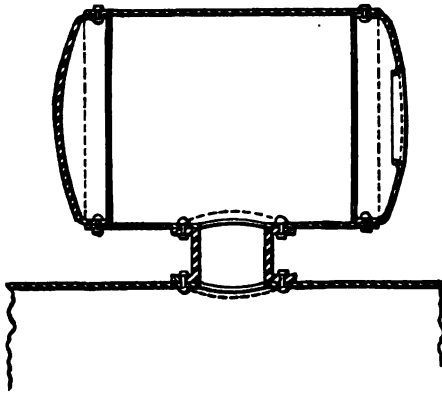


FIG. 88.—Steam-drum, Single Nozzle.

part of the surface of a sphere whose diameter is twice that of the dome. If the top be flat it will require staying. Often a manhole is placed on top of the dome, and in such cases the steam-pipe leads from the side.

A Steam-drum is better than a dome for increasing the steam-space, as it can be larger than the ordinary dome and requires less

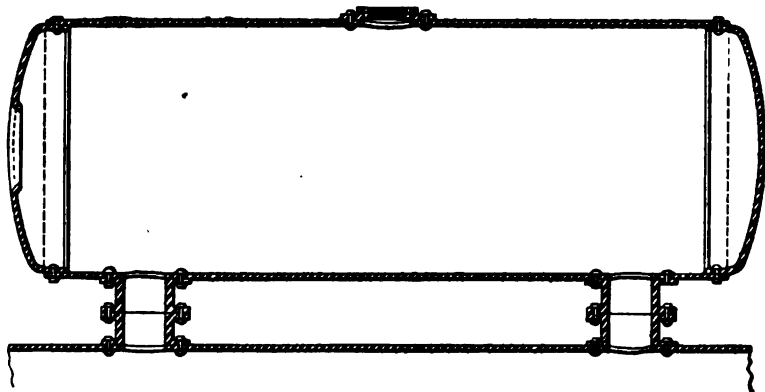


FIG. 89.—Steam-drum, Double Nozzle.

cutting of the boiler-shell (Figs. 88, 89, and 90). The drum is designed according to the same principles that apply to the shell. It consists of a cylindrical vessel having a diameter about half of that of the boiler-shell or less. The heads are most always bumped, with a manhole in one of them. It is connected to the shell by a neck or nozzle, which is made of riveted steel,

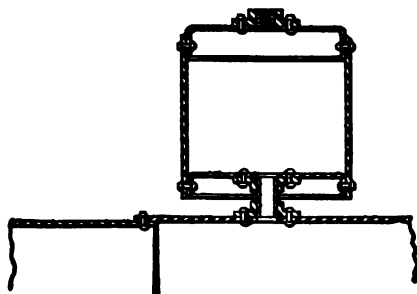


FIG. 90.—Steam-drum, Pipe Connection.

welded steel, or more commonly of cast-steel or cast-iron. When cast they are generally proportioned according to the standard sizes illustrated in Fig. 91.

The drum may be arranged as in Fig. 88, with one nozzle, or as in Fig. 89, with two nozzles. This latter method is objectionable

on the ground of unequal expansion. If the drum be so long and heavy as to require two supports, it is better to have a nozzle at one end and a false nozzle or saddle at the other.

Sometimes one drum is common to two or three boilers, and is then placed at right angles to the boiler axes; but this arrangement,

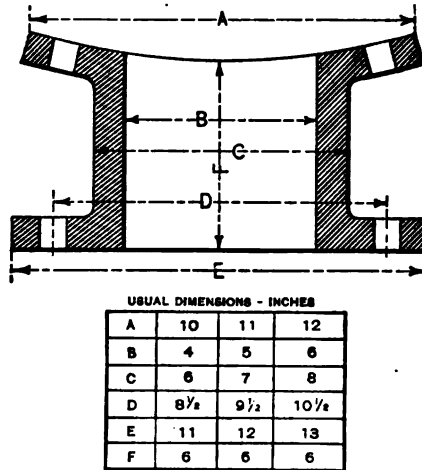


FIG. 91.—Standard Nozzles of Cast-iron or Cast-steel.

while convenient, is difficult to maintain, due to the stresses caused by differences in expansion, and prevents the use of one boiler without the other unless a stop-valve is interposed. It is, therefore, most serviceable in large batteries.

The drum forms a very essential part in the design of most water-tubular boilers.

A Steam-superheater is a large steam-drum through which a flue passes, conveying the products of combustion from the boiler to the uptake or stack. They are misnamed, as they are not designed primarily to superheat the steam, lacking sufficient heating surface to be effective. They are made in various styles, some of which are illustrated in Figs. 26 and 27. They are usually supported on the shell of the boiler in such a manner that the weight of the superheater is carried by its flue. The adoption of outside stays between the shells of superheater and boiler are not to be recommended, as they frequently are sources of trouble from unequal expansion.

The steam pipe from the boiler should enter at the side of

the superheater, although it may enter the bottom. In the former case there should be a small drain from the bottom of the superheater back to the boiler. This drain is generally made of seamless drawn copper pipe, from 2 inches to 4 inches in diameter, according to requirements, but the smaller the better, and it should be curved for expansion. This drain should enter the boiler-shell just below the water-line, and no check-valve is required unless a great difference in pressure is expected.

A manhole should be worked in at some convenient place, and usually the upper head is selected.

The flue, or liner as it is called, may be made of one flue or of a number of smaller ones as desired. These liners are proportioned according to the rules for a flue subjected to external pressure, but with a large factor of safety, as it is all superheating surface and exposed to high temperatures. The United States Steamboat Inspection Rules for liners are given under Flues, Chapter VIII.

When a heavy stop-valve is placed on the side of a superheater or steam-chimney, the joint is apt to leak and cause trouble, unless the valve be securely supported and braced. The flange-bolts do not have sufficient leverage to resist the continual vibration caused by the engine shaking the steam-pipe. The valve can often be supported by long flat or angle braces, leading from the outer flange to points higher up on the shell, as in Fig. 92.

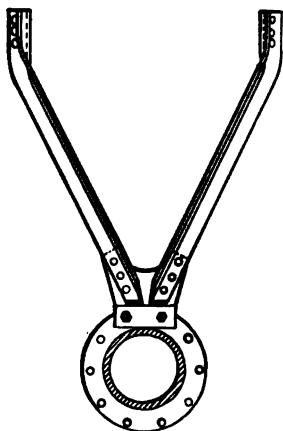


Fig. 92.—Angle Braces to Support Stop-valve.

A **Steam-chimney** is a steam-dome of large size, through which the smoke-flue passes, as in Fig. 25. It necessarily weakens the shell by cutting out so large a piece, and has all the disadvantages of the dome, without any additional advantages over the ordinary forms except that of size.

Steam-chimneys are seldom used except on "marine" type of boilers designed for steam pressures of 40 pounds on the inch or less.

Steam domes, drums, superheaters and chimneys are but make-shifts when used for increasing the steam-space, and are best avoided whenever possible.

Steam-pipes should be of such size that the average velocity of flow does not exceed about 8000 feet per minute.* The area through valves should be somewhat in excess of that of the pipe, so as not to create loss of pressure due to friction.

The steam-pipe leading from a boiler may cause priming if made too large. A rule for size of steam-pipe to suit a boiler, as stated by Seaton in Manual of Marine Engineering, is that the area should not exceed the following:

Area in square inches = $(0.25 \times \text{grate area in square feet} + 0.01 \times \text{heating surface in square feet}) \times$

$$\sqrt{\frac{100}{\text{Pressure in pounds per square inch}}}$$

In cases where a steam-drum or superheater is used, the steam-room in the boiler proper is generally small for the engine, and, further, the engine is apt to be of the early cut-off, long-stroke, slow-speed type. The greater care should then be taken to see that the pressure does not vary too much in the boiler at each gulp of steam taken by the engine. The steam-pipe between the boiler and the drum or superheater must not be made too large. It may be as large as that leading to the engine but not larger, or it may even, with good result, be made somewhat smaller. The area should not exceed that given by Seaton's rule, and, according to circumstances, should often be much less. When the steam is taken by gulps at long intervals, the drum or superheater may act like a reservoir; and changes of pressure occurring in it will cause a more or less steady flow from the boiler.

If wet steam is expected, a steam separator on the steam-pipe near the engine is recommended. A separator is a safeguard in every case, as it will prevent a possible accident to the engine from water, whether the water comes from priming, condensation, carelessness or otherwise, and it also will act as a steam-reservoir.

Pipes must be strong enough to withstand the required pressure. Wrought-iron and steel pipes are made amply strong for all reasonable pressures, on account of the thickness necessitated by conditions of manufacture. For copper pipes, the British Board of Trade rule may be safely taken as a minimum; namely, for copper steam-pipes, when brazed,

$$\text{Thickness in inches} = \frac{p \times d}{6000} + \frac{1}{16}$$

* In large pipes, the velocity may be 16,000 feet.

and when seamless, not exceeding 8 inches in diameter,

$$\text{Thickness in inches} = \frac{p \times d}{6000} + \frac{1}{32}$$

in which

p denotes working-pressure in pounds per square inch, and

d denotes inside diameter in inches.

Long bends should be made one gauge thicker than the straight parts and short bends two gauges thicker, as the material at the back is thinned by bending. The result is that the bend is of unequal thickness and more rigid than necessary, an argument in favor of as long a bend as possible.

Failures of Steam-pipes are caused by poor design, carelessness or neglect, and seldom by weakness due to the pipe having been made originally too thin. The majority of failures are traceable to lack of provision for expansion and contraction, and to the movement occasioned by the vibration of the engine. The other principal causes are lack of suitable means for drawing off the water of condensation, faulty workmanship, and defects that have developed while the pipe has been in use.

Steam-piping can be so designed that ordinary carelessness or foaming of the boilers will not cause an accident. The difference in cost between a good and a bad design is never large enough to be of any importance. Lack of sufficient head room, however, often creates considerable difficulty in laying out a steam-piping system, and occasionally prevents the designer from adopting the best arrangement.

The materials used for steam-piping are copper, wrought-iron, mild steel and cast-iron, and the design should conform to the material employed.

Copper pipes are either brazed or seamless-drawn. Seamless-drawn pipes can be made as large as 8 inches in diameter and bends can be worked by the coppersmith from straight pieces.

Brazing is now usually made on a lap seam, but formerly was done by dove-tailing the edges. When pipes are brazed, the straight pieces have one seam, and bends of small sizes can be made from such straight pieces. Bends in large pipes are made in halves with two brazed seams, one on each side.

The flanges are generally brazed on, while the end of the pipe is bent over into a recess turned in the face of the flange, to pre-

vent its being pulled out. At times the flanges are riveted in addition to the brazing.

Copper was originally adopted for pipe-making on account of its ductility and flexibility. Its extended use is now due to custom, as these properties are not found to be permanent, but are dependent upon the treatment of the material while in service. When thoroughly annealed, copper is very soft and takes a permanent set at pressures as low as 4500 pounds per square inch. Its elongation will be between 30 and 40 per cent in test-pieces 8 inches long, and its ultimate strength about 28,000 to 30,000 pounds per square inch. Under stress, often repeated, the copper will harden and gradually have its ductility decreased, but may be restored to its original condition by being annealed. It would be advisable to periodically anneal all copper steam-pipes, but this is a difficult process, as very few works are capable of annealing a full length of pipe at one heat, and when done at successive heats there is danger of leaving parts hard, thereby producing an unhomogeneous pipe which may be worse than leaving it unannealed.

Copper pipes are frequently reinforced for additional security when larger than 8 inches in diameter. The reinforcement not only strengthens the pipe, but also greatly confines the place of failure. It is done by wrapping the pipe with copper, steel or delta-metal wire, or by fitting bands of wrought-iron or other suitable material at short intervals. The diameter of the wire is generally $\frac{1}{8}$ -inch or $\frac{3}{16}$ -inch, and is stretched on under a tension of about 3000 pounds per square inch of section. It may be wound in spirals with closely laid coils, usually three wires being used for safety; or it may be shrunk on in separate bands with the ends twisted. A system of banding is shown in Fig. 93 (Marine Engineering, August, 1899). The figure is self-explanatory, and shows the method of driving the cotters, as well as of passing bends and of making a hub for a branch.

The coefficient of expansion of copper is 0.00000887 for each degree Fahrenheit.

Wrought-iron and steel pipes are now being largely used in lieu of copper. When steel is used, only the mildest qualities are selected. Wrought iron is preferred to steel by many engineers on account of the greater certainty of making a strong welded joint.

In both iron and steel pipes the joint is lap-welded and never

butt-welded, although in small sizes the pipes can be solid drawn. With steel pipes, a riveted butt-strap is sometimes fitted over the weld, but if the best material is used and care taken in making the weld, this strap is unnecessary. It then only adds additional

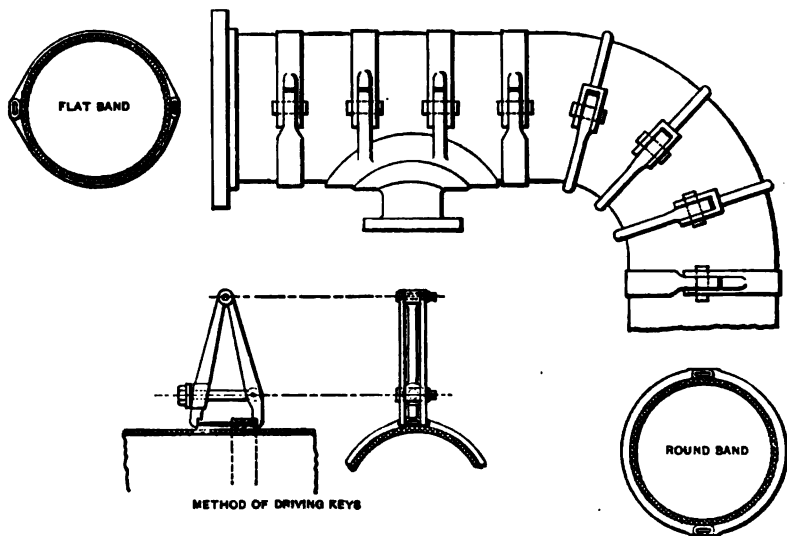


FIG. 93.—Reinforcing Steam-pipes.

cost, weight and numerous rivet-holes, which are always liable to leak and cause annoyance.

Riveted steel pipes are not used to any great extent, as they are expensive and liable to leak at the rivets and seam.*

Iron and steel pipes are nearly always made straight, but bends may be made with a radius of three times the bore for pipes less than 6 inches in diameter, and four times the bore for pipes as large as 12 inches.

The effect of corrosion of iron and steel pipes is not serious, as thus far shown by experience.

The coefficient of expansion is 0.00000648 for each degree Fahrenheit, or only two-thirds that of copper.

Cast-iron is seldom used for steam-pipes, due to its treacherous nature, but is used for flanges and fittings. The best cast-iron for pipe-making is charcoal iron with 3 per cent of aluminum to pre-

* Reference is made to paper on "Riveted Steel Pipe," with discussion, Trans. Am. Soc. Mechanical Engineers, Vol. XV, 1894.

vent blow-holes. The coefficient of expansion of cast-iron is 0.00000556 for each degree Fahrenheit.

A duplicate system is not necessary with a well-designed steam-piping plan. A system in duplicate is expensive, contains more joints and generally increases the condensation. But when used the separate systems are connected with "Ys" at the boiler and at the engine.

They are arranged according to one of three general plans, thus:

1. Two sets of mains, each of small size, but of an aggregate area to suit the plant. Both are in use, but one could be shut down for repairs while the other was operated in times of emergency.
2. Two sets of mains, one of full size and one of smaller, the smaller to be in reserve and made small to save cost.
3. Two sets of mains, each of full size, but only one in use at a time. The second main is in reserve.

The relative merit of these plans is in the order mentioned, unless some unusual condition exists. The first plan is the cheapest and strongest, as the pipes are of small diameter, and under regular working conditions there is no idle part.

In modern plants, duplicate piping is little used, and continued experience confirms this view. Some electric-lighting stations and plants of similar character still retain them, but there is a strong feeling adverse to their adoption, on the ground that they are a needless expense and increase the radiation surface, joints, valves and fittings.

Ample allowance for expansion must be provided in all steam-pipe designs, as most of the failures that have occurred have been circumferential fractures near the flanges, instead of longitudinal fractures near the middle of length as when the pipe is burst when testing. Such failures are caused by expansion strains, since cylinders are twice as strong circumferentially as longitudinally when subjected to internal pressures.

The simplest method is to provide angles in the line of piping, using fittings with easy turns, and have the legs entering the elbows long enough to take up the expansion. A better method, especially for high pressures, is to design the pipe with bends of long radii, remembering that the thicker and stiffer the material the longer should be the radius. Still another method is in vogue when angles and bands cannot be used—that is, the adoption of expansion-or

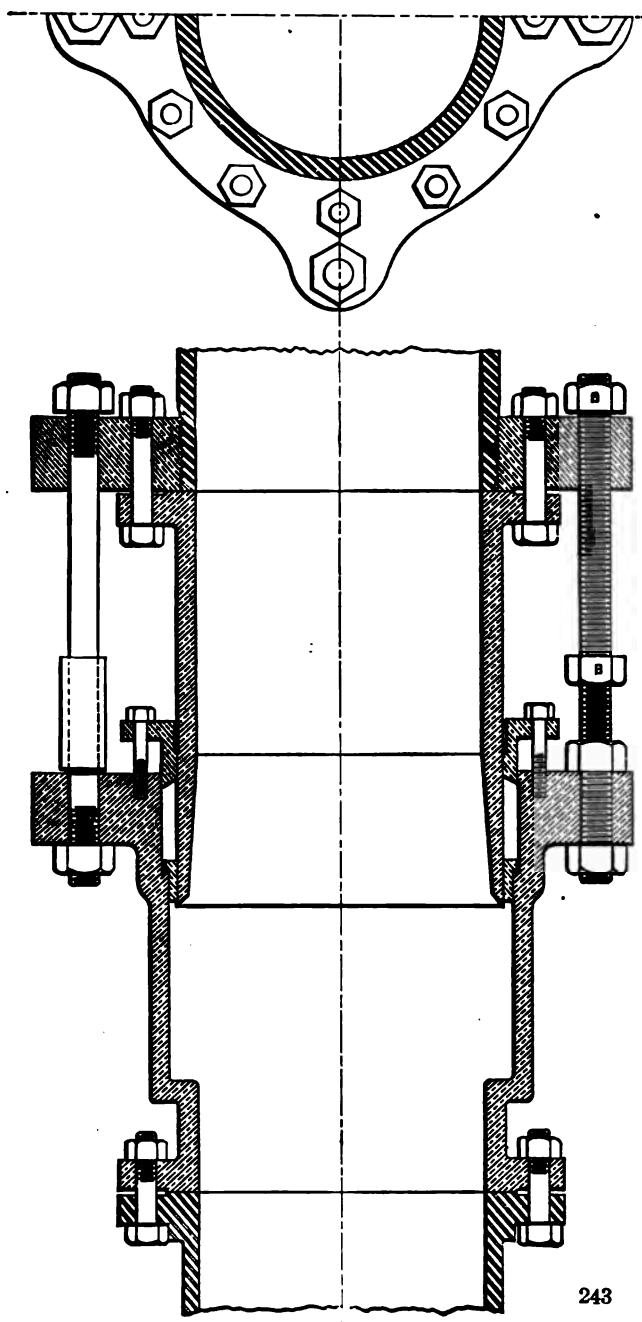
slip-joints. Copper pipes seldom require slip-joints, as the material, being both flexible and ductile, can usually be designed so as to receive the required bends. Cast-iron, wrought-iron and steel are, however, so rigid that the required length of bend cannot always be provided and slip-joints must be made. Some engineers recommend, out of preference, such straight pipes with slip-joints. A slip-joint with stuffing-box is illustrated in Fig. 94.

These joints are liable to give trouble, but with care in the design of the whole arrangement they need not be more troublesome than flanged joints subjected to cross-stresses or copper pipes hardened by repeated alteration of form.

In the design of the slip-joint it is essential that the lengthening of the pipe shall actually work into the joint, and neither enter so far as to bind nor pull out by contraction or blow out by pressure. This is accomplished by securely fastening some part of the pipe so that all expansion must take place from that point, and by securing the stuffing-box part of the joint in a fixed position. When both the joint and the far end of the entering pipe be fixed, there is no danger of the joint blowing out, unless there be a bend in the pipe. When these bends occur, safety-stays should be used to tie the bend to the fixed part. These safety-stays are frequently fitted on slip-joints with straight pipes, but are not necessary. Safety-stays, two or four, according to circumstances, are fastened to the fixed part of the joint and pass through a safety-flange on the pipe, located about 18 inches or 20 inches back from the tail pipe. On each side of this flange there is a nut on the stay. When the pipe is cold the outside nut should be carefully adjusted and fixed by pinning, so that it cannot be carelessly screwed up against the flange when the pipe is expanded by heat. The inside nut prevents the pipe from pushing too far into the stuffing-box, and should be adjusted in like manner when the pipe is hot, making due allowance for movement caused by vibrations of the engine. This inside nut may be omitted altogether, thus preventing any thoughtless tightening, or it may be replaced with a loose ferrule slipped over the stay and cut the proper length to reach between the safety-flange and the flange on the joint.

When long mains are used to carry steam at low or moderate pressures, the expansion can be provided for by designing a double offset with six elbows secured by screw-threads. With this arrangement the amount of expansion should be limited by adopting frequent

FIG. 94.—Slip-joint with Stuffing-box for Steam-pipe.



offsets and as many points of fixed support, since the expansion is taken up by the working of the threads. Such offsets will form a pocket for water which must be drained off through a trap of suitable size.

Flanges are used to join the pipe-lengths, although iron or steel pipes when less than three inches in diameter can be screwed together with couplings. These flanges should be made standard sizes, so as to be interchangeable and insure accurate fitting.

Flanges for copper pipes are made of brass or of composition, and for iron or steel pipes of the same or of cast-iron, cast-steel, malleable-iron; or wrought-iron forged into shape and welded on.

Copper-pipe flanges are brazed on, and are sometimes riveted in addition. The commonest flange is illustrated in Fig. 95, in which the flange is slightly bevelled so as to admit the brazing

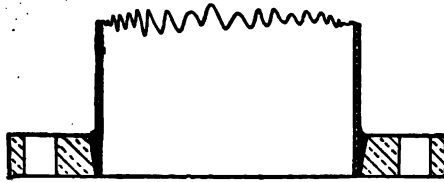


FIG. 95.—Plain Flange for Copper Pipe.

solder. It is the simplest arrangement and the most certain that the solder will be where most needed. Another form, Fig. 96, has

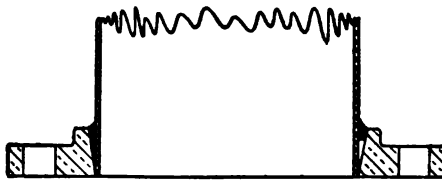


FIG. 96.—Collar Flange for Copper Pipe.

a collar designed to give additional strength, but there is danger that the solder will not enter all the way, as shown on one side, thus becoming firm only at the weak end of the collar. Another form, Fig. 97, has a bevelled sleeve over the pipe in place of the collar, but then there is danger that the solder will not run under the sleeve, as shown on one side of the figure. It is best to extend the pipe through the flange and turn the edge up into

a recess in the flange-face. The flange should then be brazed and riveted on as in Fig. 98.

On iron and steel pipes, when less than 16 inches diameter, the

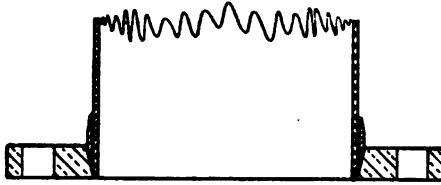


FIG. 97.—Plain Flange with Sleeve for Copper Pipe.

flange is generally screwed on; and when over 16 inches diameter it is riveted on. The objection to screwing is the tendency to leak along the thread. This tendency can be remedied by screwing

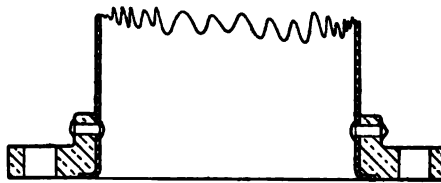


FIG. 98.—Collar Flange with Edge of Copper Pipe Turned Over.

the flange up hard on the pipe (Fig. 99) and then cutting off the projection by facing up both the flange and pipe end; or by leav-

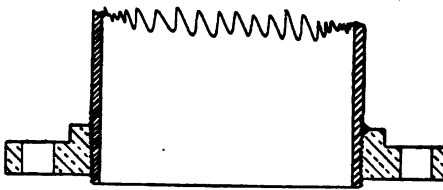


FIG. 99.—Flange for Iron or Steel Pipe.

ing a recess in the collar and calking in lead, as shown on one side of Fig. 99. Forged flanges welded on are excellent, but cost more than screwed flanges.

The faces of the flanges should be faced true so as to fit snug and tight. When the faces are plain, a gasket of rubber or other packing material is used, although ground flanges may be bolted

metal to metal and made tight. Other forms of face are used:—
Fig. 100, in which the flanges have a tongue and groove turned

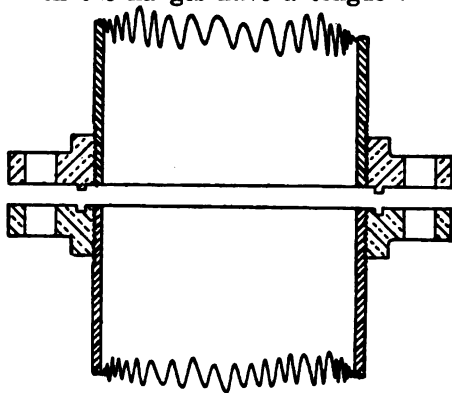


FIG. 100.—Flanges with Tongue and Groove.

on the faces and are bolted together over a copper ring as a gasket. The objection is sometimes the difficulty of removing and inserting a pipe section. Fig. 101, in which the pipe ends are flanged

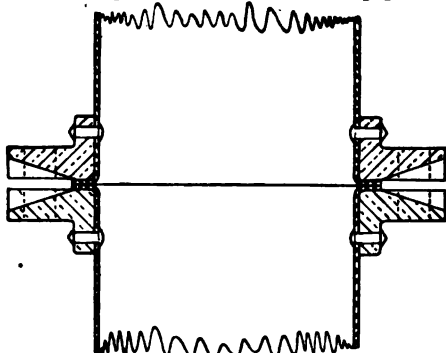


FIG. 101.—Flanges Cut Away to Facilitate Calking Edges of Pipe.

and calked. This is the Walmanco or Van Stone joint, and when used with charcoal iron or mild steel pipes the rivets are omitted. It is one of the best flanges, as it is simple and can be kept tight under heavy pressures; although only the best grades of iron or steel can be used, as poor material will not stand the flanging. Fig. 102, in which the faces are made with a recess and projection to fit into each other. This arrangement prevents the gasket from being blown out, no matter

what the pressure. Fig. 103, in which the flanges are faced and ground true and bolted directly together, an arrangement that has proved satisfactory with pressures as high as 200 pounds.

The general design of the steam-piping is all-important. Short and direct connections between boiler and engine, with

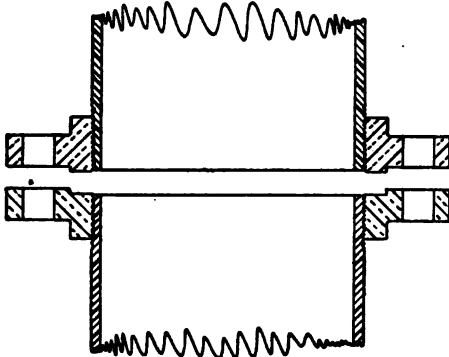


FIG. 102.—Flanges with Recess and Projection.

consideration for expansion and drainage, are to be preferred as being simple and offering least loss from condensation. Design the piping so that there will be no pockets, but if pockets must

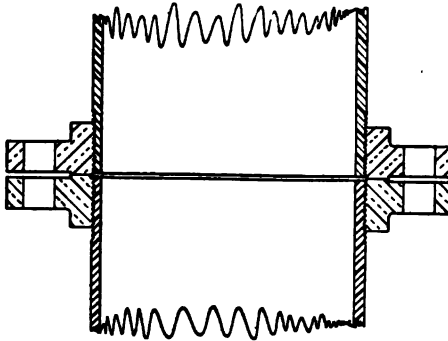


FIG. 103.—Flanges with Faces Ground to Fit.

be formed, make them as few as possible and of ample size with drains. It is an easy matter for steam flowing at 8000 feet per minute to pick up water and carry it along.

The piping should always pitch for drainage in the direction of the current of steam, and horizontal pipes should have a fall of

at least one inch in every ten feet run. Horizontal pipes of unequal diameter may be joined by an eccentric flange on the smaller pipe, so that the bottom of the pipes on the inside shall be level to facilitate drainage.

The area of the steam-pipes should be amply large for their duty. A large steam-pipe is a good fault, especially when it supplies engines not using regular quantities of steam, as it then acts as a reservoir. With very high pressures or with a regular flow of steam, too large a pipe may become dangerous, as it may serve as a lodging-place for water.

It is best to keep as much of the piping in the boiler-room as circumstances will permit and as little as possible in the engine-room, because less damage will, as a general thing, be caused there in case of an accident, and the chance for fatal injury to the attendants is diminished.

If a horizontal distributing steam-main be used, the feeder-pipes from the boilers should each rise out of its boiler with a long bend and enter the top of the main. * All supply branches also should take off from the top of the main. The main should be drained from the bottom by special drain-pipes and the feeders should never be used for this purpose, as water will not flow back against the velocity of the steam.

When these distributing mains are short, it is best to anchor them in the middle and divide the expansion between the ends. When they are long, place an expansion-joint in the middle and anchor half way between the joint and the ends.

A steam-main frequently has its area diminished as branches are taken off, and is terminated in a bend to the last engine or in a tee with two branches to the last two engines. This practice is questionable and should not be adopted with high pressures. It is better to make the main full size or nearly so for its entire length (unless the main be exceptionally long), giving it the proper pitch, say one inch in each ten feet run, and to terminate it in an elbow, turned downward with a vertical pipe attached 3 feet or 4 feet long. This pipe should be capped and a drain arranged discharging into a trap, a receiving-tank, or back to the boiler if the water will return by gravity.

All the valves on the steam-piping should be so located as not to collect water when either closed or open. When the pipe

rises from the boiler with a long bend, the stop-valve should be at the top of the bend and at some distance from the boiler connection. It is best to have two valves between the boiler and the distributing main, and double valves are sometimes required by local ordinances.

Globe valves should not be used on horizontal pipes, as they necessarily form a pocket for water. On small pipes, an offset globe valve may sometimes be used, but in all cases it is generally better to use a gate-valve with a screw movement. If the pipe is large, it is well to provide a by-pass in connection with the gate-valve.

A gate-valve should always be placed with the spindle vertical, but when head room is lacking, the valve may be placed horizontally. Gate-valves never should be set with the spindle pointing downward, as then the valve will always form a pocket, no matter what the opening may be, and the gate is liable to be damaged by water-hammer when partly open.

A stop-valve placed directly on the boiler nozzle with a vertical pipe leading from it is a very bad arrangement, as water is sure to collect. If such an arrangement cannot be avoided, place a drain on the upper side of the valve. It would be better to place an angle stop-valve on top of the vertical pipe and have the supply main properly pitched away. As stop-valves on tops of boilers are often inconvenient to reach, an elbow can be placed on the boiler nozzle and the pipe led from it with the valve located at some accessible point. This pipe should pitch from the valve in both directions. This arrangement is very good when head room is low.

All valves should be located for easy accessibility in order to facilitate rapid control. On this account it is well to group them as near together as may be convenient. Usually in large stations the valves can be so placed as to be reached from a light platform or gallery suspended from the roof trusses.

A Stop-valve should be placed on the steam-pipe, so that the steam can be shut off at any time. This valve should be close to the boiler, and is frequently mounted on a nozzle attached to the boiler shell, although its best location must depend on the piping design. The valve should be operated by a screw, and when of the globe pattern be arranged to close down against the pressure,

that it may be repacked around the spindle when required. It never should be a quick-opening valve of the gate or lever type.

Stop-valves are generally made of cast-iron, with the valve, seat and spindle of gun-metal or bronze. The best valves are made all of bronze, but this is a refinement little adopted except in naval vessels. In very important work, and in some war vessels, the stop-valve is made to close automatically, whenever the pressure in the boiler falls below that in the steam-main, and to automatically open again when the requisite pressure has been regained. This is done to prevent total shut-down in cases of damage to one boiler, when a battery of boilers are feeding into a common main.

Since bronze and cast-iron have different coefficients of expansion, stop-valves often leak at the seat unless the seats be well secured into the cast-iron. If the valve be fitted with wings to guide its motion, these wings should be curved slightly, so as not to bind against the seat except when the valve is down.

The area through the stop-valve should exceed that of the steam-pipe, so as not to cause loss of pressure due to friction.

In all globe and angle valves the full pressure is on the valve when closed, and the load is carried by the yoke or bridge and by the spindle. These parts, therefore, should be amply strong. The rule for size of spindle in Seaton's Manual of Marine Engineering is:

$$\text{Dia. of spindle} = \frac{\text{dia. of valve}}{50} \times \sqrt{\text{pressure}} + \frac{1}{8} \text{ inch.}$$

It is well to have the yoke so designed that it will carry a spindle for regrinding the seat when necessary.

The drips from all steam-mains should lead to a receiving-tank, and the water be automatically pumped back to the boiler. If the drips lead from mains under different pressures, then the drips should be trapped. Sometimes the drips can be drained back by gravity into the boilers through a check-valve at the boiler, but the receiving-tank system is generally the better.

All condensed steam containing grease or oil should be kept separate and passed through a grease extractor or be filtered before being returned to the boilers.

The Dry-pipe, sometimes called the "internal pipe," is an

arrangement for drawing steam from all parts of the steam space, and is a very good device for obtaining dry steam and for preventing priming. It consists of a pipe fitted inside of the boiler and connected to the steam-main (Figs. 11, 16, 17, 18, 19 and 20). This pipe is perforated with holes or slots on its upper side only, the ends being closed. There should be $\frac{1}{2}$ -inch or $\frac{3}{4}$ -inch holes in the bottom near the ends to act as drains. The aggregate area of the holes for steam entrance into the dry-pipe should be about equal to that of the steam-main.

The dry-pipe is frequently made as a trough or half pipe, the top of the boiler shell forming the cover (Figs. 12, 26 and 27). The edges of the trough are kept about one-half inch from the shell, so as to form a passage for the steam.

The dry-pipe operates by drawing steam from the length of the steam space, and therefore renders the separation of the steam from the water more uniform and prevents local differences in pressure, which cause uneven ebullition and priming.

Dry-pipes may be made of cast-iron, which is the cheapest material and obviates any chance of galvanic action; or of wrought-iron or steel, which are both light and serviceable; or of brass or copper, which are the most expensive and most easily worked, but objected to by engineers as tending toward galvanic action. Copper pipes should be tinned.

The dry-pipes are held in place by lugs or bands bolted to the shell.

Boiler-feed. Every boiler should have two independent means of feeding in the water, and marine boilers subject to inspection are required to be so fitted. Many stationary boilers, however, have only one feeding attachment.

Each feed-pipe entering the boiler should have its own back-pressure or check-valve, so situated that the check will fall by gravity into the closed position, and it should be placed as near the boiler as possible, with a good screw-closing stop-valve between it and the boiler. This latter valve is used for shutting off the connections to affect repairs to the check-valve or piping.

The feed-pipes should be of copper or brass, as those metals are the most durable and present the neatest appearance. All valves on the feed-pipes may be of the globe pattern, but good gate-valves are better, except for the stop-valve, as they offer less

frictional resistance and make a more direct passage for the water. For similar reasons all elbows and bends should be of the long radius type.

If any of the feed-water comes from condensed steam containing oil or grease, it should be filtered, as also any water that is muddy or carries sediment in suspension.

There is some diversity of opinion as to the best place to admit the feed. When the feed is very hot, there can be less difference in actual evaporation, no matter at what point it enters. Unfortunately the feed-water is often cold, or rather much colder than the water in the boiler, and then it would appear best to so admit it that it will assist the natural circulation and at a place where it will never strike directly against hot heating surfaces, even when the feed intended for a battery of boilers be concentrated into one by accident or otherwise.

The best place can only be determined after careful study of the type of boiler and of the scale-making qualities of the water. Many prefer to have it enter near the bottom, some at the coldest part, while others favor its admission near the water-line and at some place where there is a natural downward current.

The consensus of opinion is to admit the feed near the water-line, and to distribute the discharge through a number of small openings, so as to prevent a solid jet of more or less force from entering. If the pipe enters near the water-line and turns down on the inside, care must be taken that the open end should be always in the steam space or below the lowest water-line, and never be covered and exposed alternately by possible variations in the water-level. If the feed-pipe fills with steam, loud explosive noises will result as condensation takes place. These explosions, while not dangerous, are extremely unpleasant and are apt to cause injury and leaks in the joints. Such action sometimes occurs in improperly piped marine boilers, which are subject to wide variation of water-level on account of the rolling and pitching of the ship.

A very good plan is to carry the pipe into the steam space, and to terminate it in a horizontal branch, just above the water-line, with holes so that the feed will enter as spray. When the steam space is contracted, it would be better to place the distributing pipe just below the water-line. If the water carries

much lime or magnesia, it would be better to shorten the internal arrangement, leave the ends of the pipe open and omit the holes.

Another good plan, especially with bad water, is to let the feed enter an open trough carried inside the boiler just above the water-line. As the trough fills, the water will spill over the sides. In place of the trough, a disk or inverted cone may be used. This trough or cone will distribute the feed, heat it and retain considerable deposit which can be removed without injury to the boiler.

The disadvantage of any internal piping is the danger of derangement due to scale formation within and the consequent closing of the pipe. The advantage is that the feed has a chance

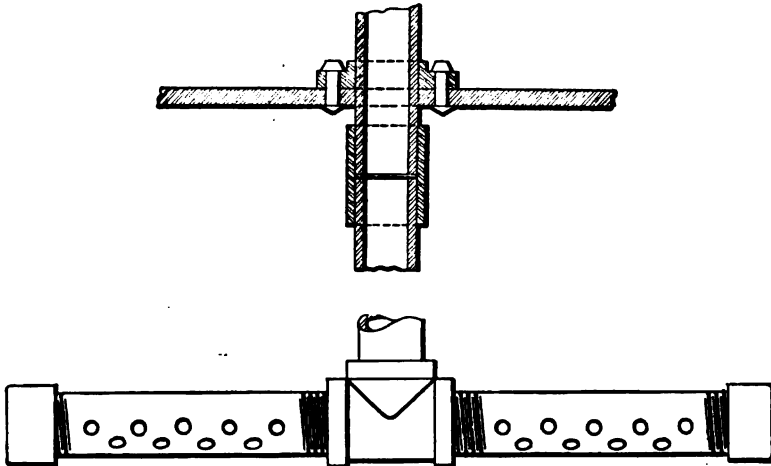


FIG. 104.—Feed-pipe Entrance with Distributing End.

to become heated before it enters the boiler. An important advantage in having the feed-pipe enter near the water-line is that the water in the boiler cannot be blown out should the check-valve fail to work from any cause.

No matter where the feed enters, it should be fed in continuously and in just sufficient quantity to equalize the water evaporated, thus maintaining a constant water-level. The practice of intermittent feed—that is, permitting the water to fall a certain amount before refilling—is objectionable and uneconomical.

Figs. 11, 12, 15, 17, 18, 104, 105 and 106 show various ways in common use for feed-pipes entering boilers.

The feed-water should be as hot as possible, both for the sake of economy and for preventing local contraction stresses caused by

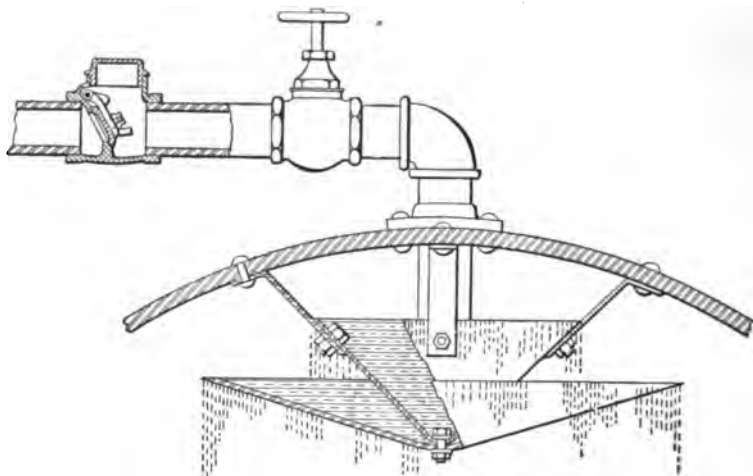


FIG. 105.—Feed-pipe Entrance.

the feed striking hot surfaces. It is forced into the boiler by injectors or pumps. Many boilers have two injectors, others two

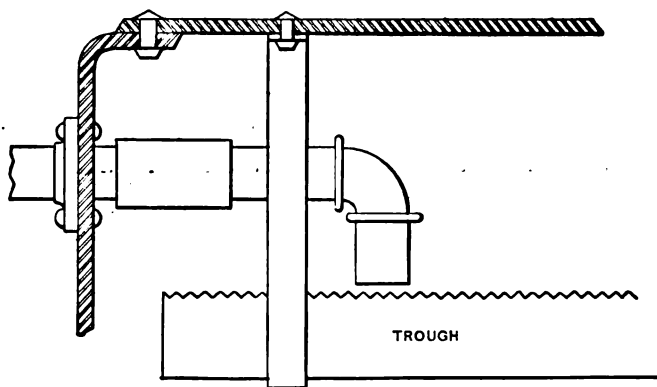


FIG. 106.—Feed-pipe Entrance.

pumps, and some one injector and one pump, so as to provide duplicate systems.

Injectors or Inspirators are manufactured in standard sizes and merely have to be piped or connected (Fig. 107).

Injectors are seldom arranged to feed more than one boiler. They are capable of lifting the supply water to a height of about 25 feet, the height depending upon the steam pressure; but with a high lift their action is not always reliable. For ordinary conditions, the lift should not exceed five or six feet.

The supply may be under pressure, but it will be found much better to use a tank with a supply pipe fitted with a "ball cock," and arrange the injector to lift the water from it.

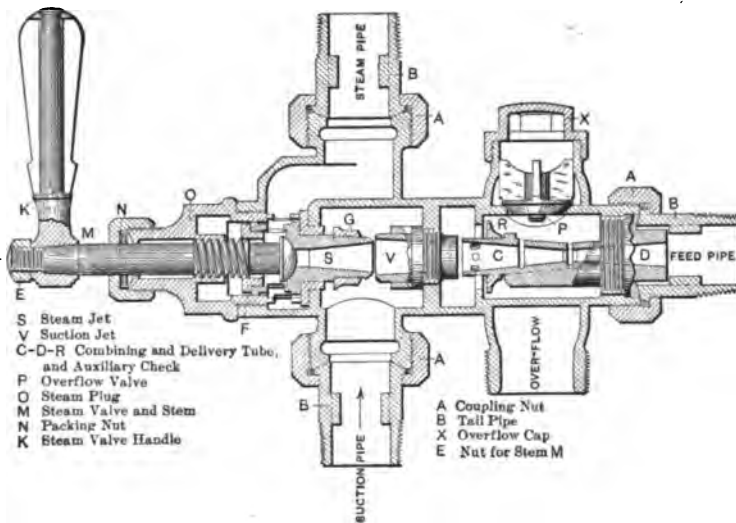


FIG. 107.—The Metropolitan Injector.

The steam-pipe connection should lead from the steam space of the boiler, so as not to be shut off by the main stop-valve. This pipe should have a valve near the injector to control the steam-supply; and it is best to place another valve near the boiler, which can be closed in case of accident to the piping. The other connections are to the water-supply, the boiler-feed and the overflow, all of which should be as direct as possible.

The principle upon which these ingenious devices operate is as follows: Steam under pressure flows through a free opening at a greater velocity than water under the same pressure. When the steam is turned on, it issues through a nozzle, sucks up the supply water, condenses and discharges through the overflow. The impact of the steam on the water gives to the latter much of

its velocity. The momentum of this water is sufficient to raise the check-valve and enter the boiler against the steam pressure, as soon as the overflow has been closed. Sometimes the steam will blow back through the suction or water-supply pipe, rather than enter the boiler, a condition generally made apparent by the noise. When such action occurs, the injector must be shut down by closing off the steam and started afresh.

Injectors may be operated with exhaust steam from an engine if at sufficient pressure, but then the steam passages must be correspondingly larger.

The injector is really a heat-engine without moving parts, the energy stored in the steam being converted into work by placing a column of water into motion, and by overcoming the frictional resistances. In order to accommodate itself to variations in pressure of steam and temperature of feed and supply water, the apparatus is so made as to adjust the openings for steam and water.

The injector has the virtue of heating the feed, but if the water-supply be so hot as not to condense the steam, it will not operate against the boiler pressure.

As a pumping apparatus, an injector is very inefficient; but as a boiler-feeder, its efficiency is high, since the heat of the steam, except the small amount lost by radiation, re-enters the boiler.

Feed-pumps are purchased from the makers in standard sizes to suit. They are nearly always made double-acting, but may be single or duplex, horizontal or vertical. Duplex pumps are more reliable than single pumps, as they are not so apt to catch and stop, but are more expensive. Vertical pumps are, in general, better than horizontal ones, although for ordinary boiler-feeding purposes, there is little choice. For marine work the vertical pump is much in favor, as it can be easily fastened to a bulkhead and thus be kept off the floor.

Feed-pumps should always have a capacity in excess of the ordinary or average requirement. One pump may feed a battery of boilers, and when so arranged the water is pumped into a supply main extending along the fronts of the boilers, from which branches are led to each boiler. The supply to each is controlled by a valve on the branch pipe.

The steam-pipe to the pump should lead directly from the boiler, so as to be operative at times when the main stop-valve

is closed. When the supply water is hot the pump should fill by gravity and not by suction, as in the latter case the hot water is liable to vaporize under the vacuum and the pump may fail to lift the water.

When the engine-room is separate from the boiler-room, it is sometimes a question whether the feed-pump should be in the engine-room directly under the control of the engineer, or be in the fire-room and exposed to the dirt and dust. Other things being equal, it is best to place it in the boiler-room under the management of the boiler attendant, and only retain an attendant who is trustworthy and competent to maintain a proper water-level.

The feed-pump may be connected to and driven by the main engine, a practice which is common in marine work and with engines that operate continuously for long periods at comparatively steady power. It is more economical to operate a feed-pump by the main engine, as the consumption of steam per horse-power is much less in the main engine than in any form of independent pump. This practice, however, is not general, because the independent pump is always available for use or repair, whether the main engine is running or stopped.

Feed-water Heaters. The feed-water should always be as hot as possible, since less heat will be required to evaporate it, and since there is less danger of injury from local contraction caused by impact of cold water. A good feed-water heater is a very desirable adjunct to a boiler plant.

The saving effected by a hot feed is considerable, and when this can be accomplished at a cost (including interest, depreciation, maintenance and repairs) low enough to establish a net saving, a heater should always be installed. The gross saving in heat-units can be computed with the aid of a steam-table, or from the formula for the total heat of evaporation (see Chapter I). The tables that are usually published showing the saving effected, give the gross saving and do not consider the items of cost mentioned above.

The water may be heated in various ways. The exhaust steam may be condensed in a surface condenser and be pumped back to the boiler. The water may be made to pass through a heater, which can be warmed by exhaust steam, by live steam taken direct from

the boiler or by the hot gases in the stack. Circumstances must determine the proper method to adopt. Live steam will cost more to produce than the saving in the heater, on account of loss from radiation, but it may pay through prolonged life of the boiler and the lessened cost of removing scale.

The surface condenser is a vessel into which the exhaust steam enters and is condensed by air or water, the condensed steam being made available for boiler-feeding. As the condensed steam contains the cylinder oil carried over from the engine, it should be filtered before being returned to the boiler. Surface condensers are used in marine work, as they save the fresh water, and are suitable for stationary practice when water is expensive or of bad quality. Air is seldom used as a cooling medium, due to its high specific heat. When water for circulating or cooling purposes is scarce or expensive, cooling towers or shallow tanks can be used, in which the same water is cooled by air, so that it can be made available again for condensing purposes.

Heaters may be of the "open" or "closed" type. In the former the feed-water is sucked through the heater and pumped hot into the boiler; while in the latter the water is pumped through the heater under boiler pressure. A heater should be easy to clean, and be arranged with a by-pass so that it can be cut out of the system for cleaning or repairs.

The ordinary closed heater of the Goubert type (Fig. 108) consists of a shell containing straight tubes. The feed passes through these tubes and is heated by the exhaust (or live) steam that is allowed to enter the shell. The usual form of connection has the steam entrance near the bottom, and in such cases air sometimes will collect in the upper part of the shell and be difficult to remove even if an automatic air relief valve be fitted. A better plan is to let the steam enter near the top and be drawn off as usual at the bottom, with the double connection joined by a by-pass pipe and valves. This method, however, is more expensive. This class of heater should not be used with waters containing large quantities of the salts of lime and magnesia. The Berryman heater contains U-shaped tubes through which the steam circulates.

With all scale-forming waters, heaters of the type of the Hoppes Combined Feed-water Heater and Purifier (Fig. 109) are excellent.

This heater consists of a cylindrical shell containing trays, one above the other. The feed enters at the top and drips from tray to tray, depositing its scale in the process, and at the same time is heated by steam under boiler pressure. The trays can be removed and the scale cleaned off.

Open heaters of the Cochran type (Fig. 110) consist of a vessel through which the feed falls as spray, and is warmed by exhaust (or live) steam. The heater is in reality closed by a back-pressure valve, so set as not to maintain an excessive back pressure on the engine and to relieve the surplus of steam. When improperly made or set, open heaters may flood and choke the exhaust, and therefore should be fitted with an automatic control valve to regulate the cold-water supply. Open heaters are easier to clean than closed heaters, but as the feed comes in contact with the steam, they should be equipped with a grease and oil extractor.

A heater may be placed in the smoke-flue, and be warmed by the escaping gases of combustion. Such an ar-

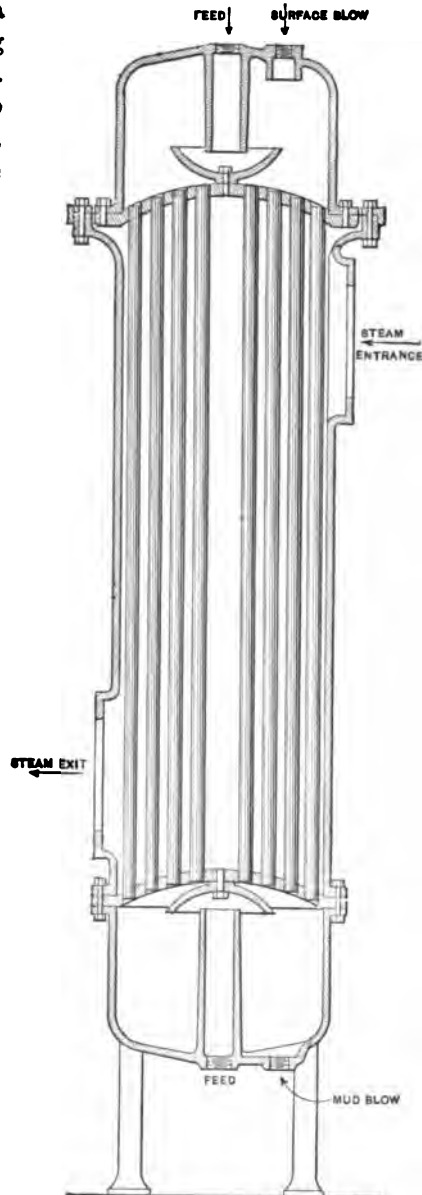


FIG. 108.—The Goubert Feed-water Heater—Closed Type.

rangement is usually called an "Economizer." The "Green" type of economizer is the most common, consisting of a set of cast-iron pipes about 4 inches in diameter and 8 or 10 feet long, surrounded by scrapers to keep the external surfaces free from soot (Fig. 111). These scrapers are worked by chains from the outside of the flue. This class of heater should be made accessible, and this can best be accomplished by dampers and a by-pass flue.

Such heaters must be made amply strong, as steam is liable to form in them. Being placed in the flue, they offer a resistance to the draft and cool the gases, so that they operate most advantageously when the draft is sufficiently strong. The reduced temperature of chimney gases and the resistance or friction to their passage diminish the intensity of the draft. Their economy lies in the fact that they recover some of the heat passing up the stack.

There is a question with natural-draft plants, whether it would not be better to have the heater surface in the boiler itself by having a high ratio of heating surface to grate surface and thus reducing the temperature of the escaping gases to a minimum.*

The first cost of an economizer is about equal to that for the same amount of heating surface, but economizer surface costs less to keep in repair and no more to keep clean than boiler heating surface of equal area. In general, all plants would be benefited by an economizer when properly designed, except very small plants, or those in which simplicity is of more importance than economy, or those in which the boiler power is ample and the gases escape at very low temperatures. Economizers have an advantage in the reserve of a large body of hot water for use in a sudden call for steam.

For the improvement of uneconomical plants or for reducing the wear due to cold feed, they are highly efficient; but with waters of bad scaling properties, heating-coils in the flue are troublesome. For such hard waters, some form of closed heater designed to take care of the scale, like the Hoppes, is better.

The tube surface of a heater of the Berryman type is generally taken from $\frac{1}{3}$ to $\frac{1}{2}$ -square foot for each boiler horse-power, while that of an economizer is usually from $\frac{1}{3}$ to $\frac{1}{2}$ of the heating surface of the boiler.

* This minimum must be, however, higher than that of the steam in the boiler, in order to obtain any effect from the last heating surface.

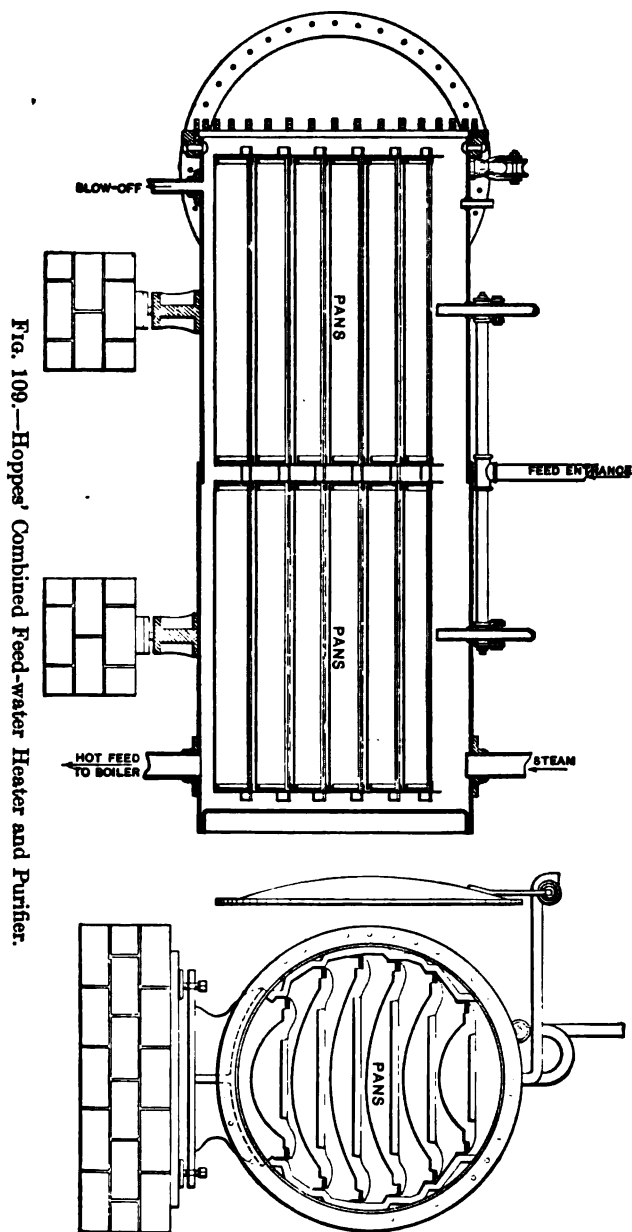


Fig. 109.—Hoppe's Combined Feed-water Heater and Purifier.

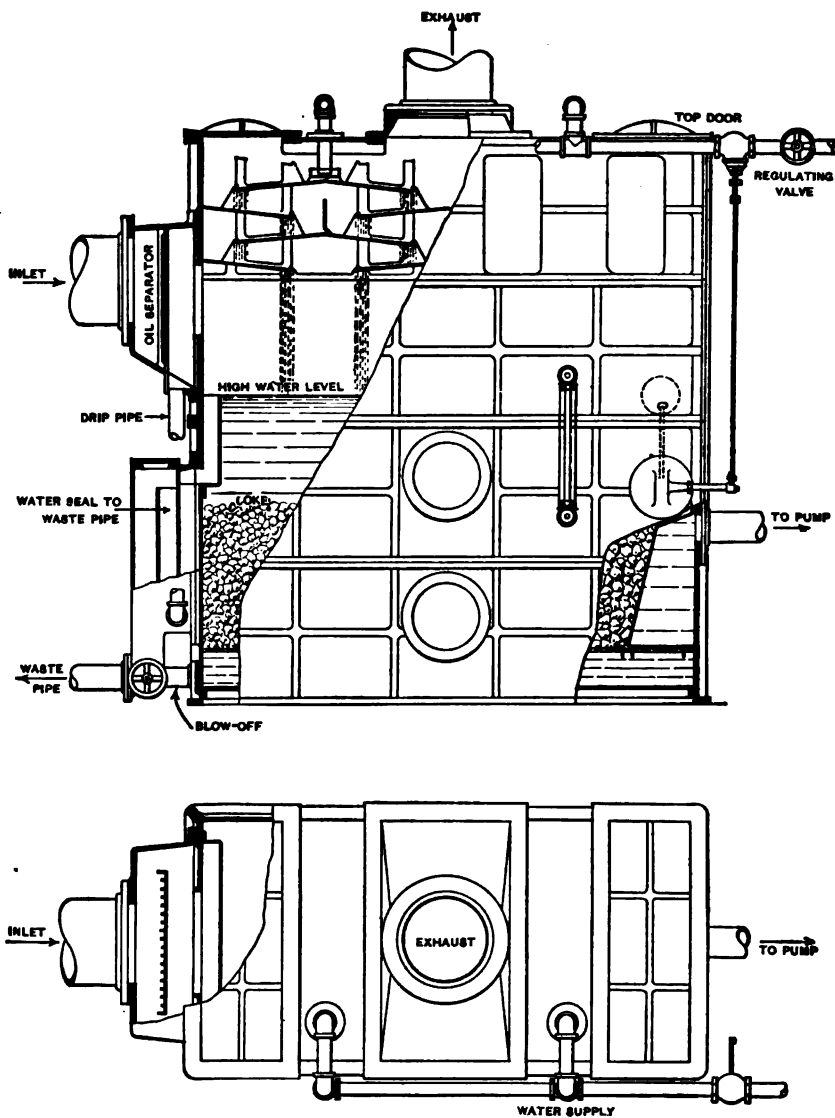


FIG. 110.—Cochran Feed-water Heater—Open Type.

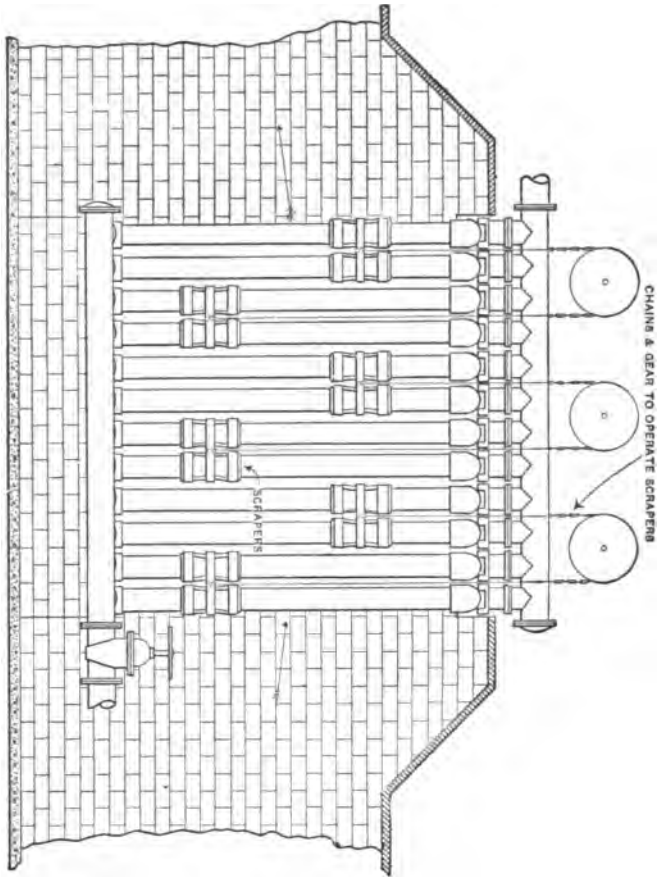
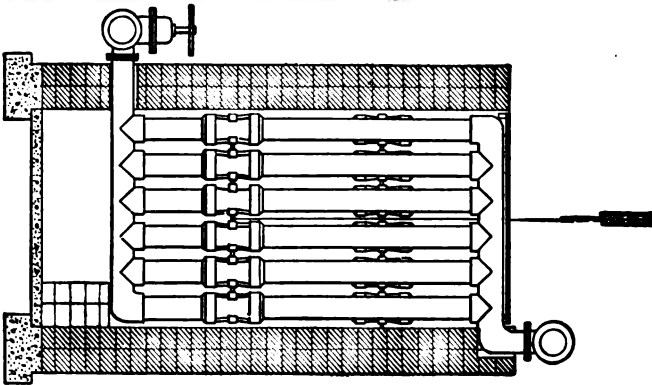


FIG. 111.—Green Economiser.



Filters are arranged in many different ways, according to the specific object desired. When sediment or sand is to be removed some good form of sand, gravel or charcoal filter-bed is generally to be preferred, but must be arranged for easy cleaning.

Closed feed-water heaters designed to catch the deposit or scale formation act in a sense as filters or purifiers.

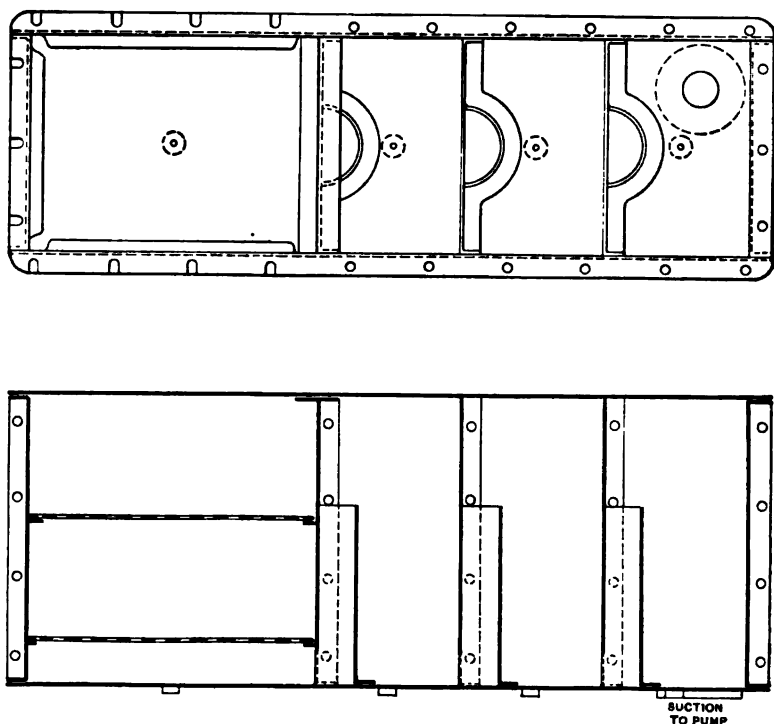


FIG. 112.—Hot-well Filter-box.

When oil or grease is to be removed, as in the filtration of condensed steam from a surface condenser, the simplest form is to allow the discharge from the air-pump to pass into a hot well or box divided by partitions. In these compartments is placed some filtering substance, as sponges, hay, salt meadow grass, excelsior (wood fibre), etc. The filtered water or suction to the boiler feed-pump is taken from the last compartment (see Fig. 112). The filtering material is removed for cleaning and may be used again.

The water may be filtered by being passed through canton-

flannel or some similar substance, which can be arranged so as to be renewed. The simplest of these filters are of the Edmiston type, and consist of a perforated cylinder over which the canton-flannel bag is stretched. The apparatus is enclosed and connected on the feed-pipe between the feed-pump and the boiler check-valve. There should be a by-pass, to use when the filter is being cleaned (Figs. 113 and 114).

Mud-drums consist of a closed cylindrical shell attached to the lowest part of a boiler and into which the feed-water is pumped. The mud, sediment, etc., in the feed-water is supposed to settle in the drum before the feed passes into the boiler proper (Fig. 115). They are not in universal use as their efficiency is doubtful, except with water containing a heavy sediment in mechanical suspension and with boilers in which the convection currents are not very rapid.

Owing to the deposit in the mud-drums, they should not be placed so as to receive the direct heat of the fire. This precaution is not always taken and then the drums are liable to burn and may be a source of danger. The drums are made of cast-iron or steel, and incidentally they act as feed-water heaters.

Water-tubular boilers are frequently fitted with mud-drums which are worked in as part of the design.

With bad waters it would be preferable to use some suitable form of filter rather than any form of mud-drum.

Blow-offs. Every boiler must have a blow-off to discharge the water. There should be two blow-offs in all important boilers, one known as the "bottom blow" and the other as the "surface

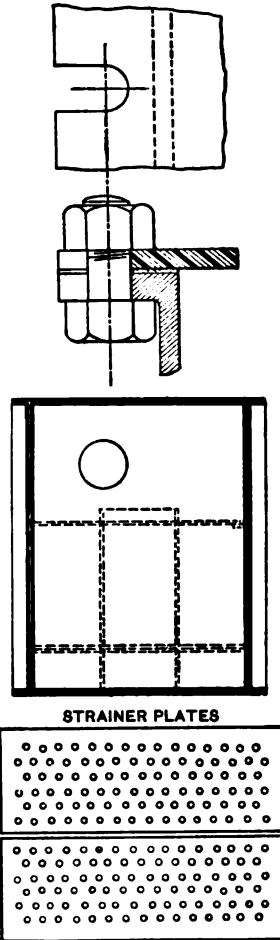


FIG. 112a.—Details of
Fig. 112.

blow." The function of the latter is to remove the scum, grease and light particles of dirt or precipitate that float or are carried

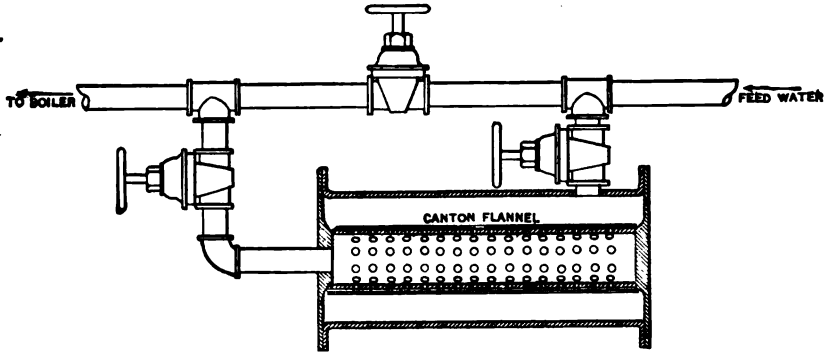


FIG. 113.—Edmiston Type of Feed-water Filter.

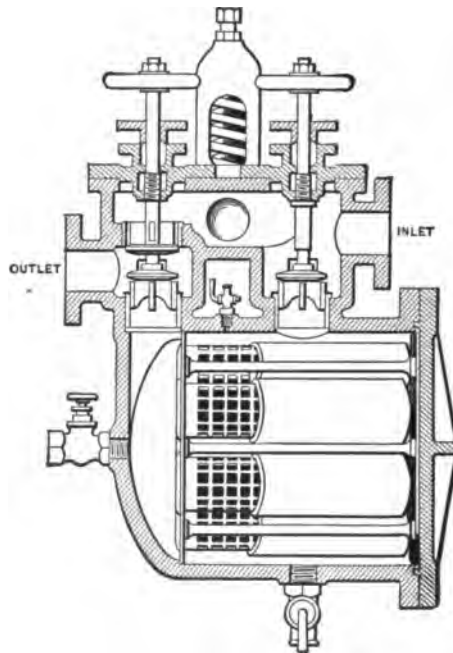


FIG. 114.—Rankine's Patent Feed-water Filter.

in suspension by the convection currents; and of the former, to remove the mud, sludge and heavy sediment that settles to the bottom.

The surface blow-off pipe is usually made bell-mouthed, with the end placed at the working water-level. The opening of the bell should face a surface current, that the water may have a natural tendency to flow into the pipe. There should be a good valve on the discharge pipe as near the boiler as possible.

The bottom blow-off pipe should enter at or near the bottom of the boiler. If it is not convenient to have it enter through the bottom, then an internal pipe should be fitted and carried down to the desired point. This internal pipe may be of iron or copper,

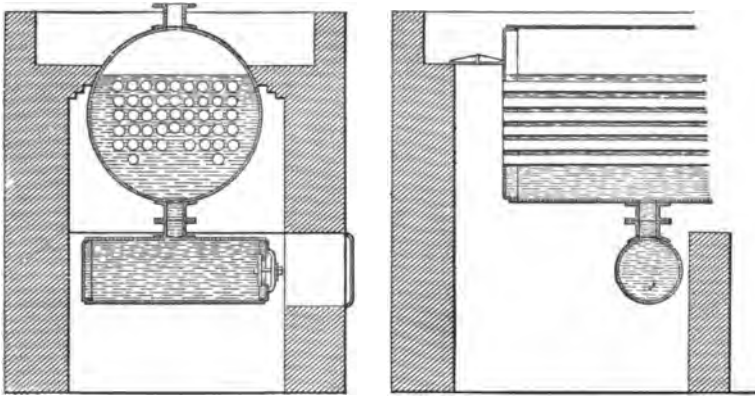


FIG. 115.—Mud-drum.

although copper may induce galvanic action unless tinned. When used to discharge mud or sludge, the internal pipe may be extended along the bottom and be perforated so as to draw from all parts; but if the water is of the bad scaling kind, such a pipe will be liable to become choked. On the discharge, there should be a valve placed as near the boiler as possible. This valve should be of the straight-way, taper-plug pattern, as such cocks can be plainly seen to be open or shut, and only carelessness will leave them partly open. A screw-valve might have a chip of scale under it, although the attendant may think it closed. A good type of valve is of the plug variety, with a gland around the stem and asbestos-packed. Taper-plug cocks are apt to corrode and stick. This can be overcome by daily use and by making them of composition. A taper of 1 in 6 is found satisfactory, and to prevent uneven wear they can be made to make a complete turn.

The discharge from the bottom blow always should be made

a sight discharge, so that failure to close the valve will be at once apparent. An invisible discharge is always attended with danger, since the water might all leave the boiler and not be noticed until too late. Furthermore, the blow-off discharges should be independent from each boiler and not be connected together.

The bottom blow-off pipes are often exposed to the action of the hot gases, and, as they may fill with scale and have no circulation in them, they are then liable to burn. They should be protected from the hot gases by placing lengths of pipe (cast-iron or tile*) around them. Never set them in brickwork where they cannot be inspected and where dampness may corrode them unnoticed. By connecting a circulating-pipe on the boiler side of the valve, a constant circulation of water may be maintained, with good result as regards overheating and durability.

The surface blow-off is generally made from 1 inch to 2 inches in diameter, and the bottom blow-off from 1½ inches to 3 inches, according to circumstances.

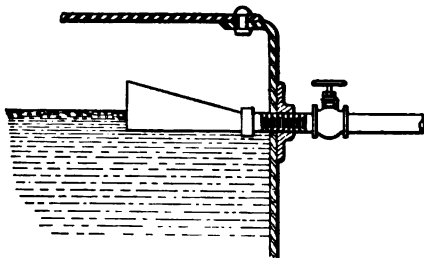


FIG. 116.—Surface Blow-off.

Boilers should never be blown off at pressures exceeding four or five pounds of steam, as serious injury can be caused by sudden cooling. Repeated short blows at frequent intervals, however, are recommended.

When boilers are located in cities, the discharge should enter a cooling-tank before it passes to the sewer, but this tank or sump should be arranged so that a pressure cannot be maintained in it under any possibility. Serious accidents have been reported from this neglect.

A surface blow is illustrated in Fig. 116, and bottom blows in Figs. 10, 11 and 12.

* Fire-clay tile having alternate ends made to fit with a recess and projection. See Fig. 12.

Safety-valve. Every boiler should have a safety-valve, and when the required size of the valve is large, it is best to use two or even three smaller ones having an equivalent area. Safety-valves are liable to stick fast on their seats, and should be raised by hand at least once every day. Safety-valves should be simple and of a type not easily deranged.

Internally weighted valves cannot be approved, as the weights are apt to fall off without warning, the rods to become corroded or be bent by a careless workman when cleaning or scaling the boiler. Such valves, not being in plain sight, cannot be readily inspected.

The types of valves chiefly used are either those externally loaded with dead weight, those loaded by a weight on the arm of a lever, or those loaded by a spring.

Dead-weight valves of the "Coburn" type (Fig. 117) have the

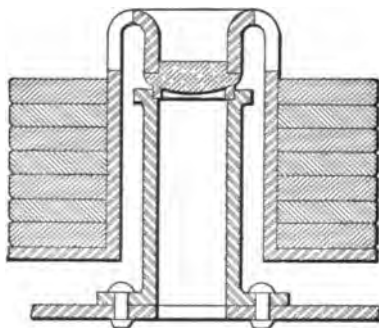


FIG. 117.—Dead-weight Safety-valve, Coburn Type.

advantage of great simplicity and can be least affected by tampering, since they require so much weight that any additional amount to seriously overload them can be quickly detected. The high pressure now in vogue, however, makes this class of valve very cumbersome. They are more used in Europe than in America.

Lever-valves are loaded by a weight carried on the end of an arm. By altering the position of the weight, a greater or less load can be easily thrown on the valve. Such valves are simple and reliable, but are more easily tampered with than the former, although experience has proved that such tampering is exceptional. Care should be taken with lever-valves to arrange the point of

support on the valve and the fulcrum around which the arm turns, in exactly the same horizontal plane, so that there may be no side pressure brought on the valve.

The problem of determining the position of the weight for any desired pressure, is one of simple proportion. The load on the valve is equal to the steam pressure times the area of valve. Neglecting the weight of the lever, this load is to the weight inversely as their distances from the fixed point of support.

Spring-loaded valves are now largely used for all kinds of boilers and are the only type adopted for marine and locomotive use, since they are independent of gravity. This class of valve can be locked so as to prevent tampering, and also can be easily operated by hand. Taking everything into consideration, a good spring-loaded valve is probably the most reliable of all the different types. They are the most expensive. Spring valves are furnished with lugs, to use when the boiler is under heavy test pressures, in order that the spring may not be overstrained. The resistance to the lift of the valve due to the compression of the spring is overcome by making the valve overlap the seat. When the valve has lifted, the steam then acts on the full area. Spring valves made in this way lift higher than ordinary conical valves.

The area of a safety-valve is the area exposed to the steam when the valve is shut, and their commercial rating is based on this area.

Valves are seldom used over 4 inches in diameter. The usual rule for size of valve required is, for dead-weight or lever-valves, 1 square inch of valve to every 2 feet of grate area; and for spring-loaded valves, 1 square inch of valve to every 3 square feet of grate.

The actual area of opening is always much less than that of the valve, and the greater the pressure the less will be the valve lift. When the seat is coned or V-shaped, the opening is less than the lift times the circumference, since the seat is oblique to the lift.

The opening should be of sufficient size to discharge all the steam that the boiler can generate. Assuming that each pound of coal can evaporate 10 pounds of water, and let

W denote the pounds of steam generated per second,

g " " grate area in square feet,

c " " quickest rate of combustion per square foot of grate per hour,

then

$$W = \frac{g \times c \times 10}{3600} = \frac{g \times c}{360} \quad \dots \dots \dots (a)$$

The quantity of steam that will escape into the atmosphere under pressures usually obtained in modern practice may be estimated thus:

Let w denote the weight of steam in pounds that escapes per second per square inch of area of opening,

P " " absolute boiler pressure,

then

$$w = \frac{P}{70} \quad \dots \dots \dots (b)$$

Valves seldom lift as much as $\frac{1}{16}$ -inch from the seat, even when made in the most approved manner. Ordinary conical valves cannot be relied upon to lift more than about $\frac{1}{8}$ -inch, and under many conditions not so much. When the seat is conical, as is the most common, the area of opening is equal to the circumference of inner edge of seat times the lift times the cosine of the angle to which the seat is bevelled.

To find the area of valve required, divide the value of W in equation (a) by that of w in equation (b), and the quotient will be the area of opening required. Divide this area by the assumed lift times the cosine of the angle of bevel of seat, and the quotient will be the circumference of valve. The corresponding area will be the valve area required.

A safety-valve does not close until the pressure has fallen somewhat below that at which it opened or "popped." This difference usually exceeds four pounds.

Safety-valve casings are generally made of cast-iron, although they are sometimes made of gun-metal. The valve is of gun-metal or brass, and the seat of gun-metal or of nickel screwed into a gun-metal base.

In locating a safety-valve, always place it as close to the boiler as possible and avoid all liability of interference with the stoppage of the connecting pipe. It is best to place the valve at the boiler and carry away the escaping steam through an open pipe rather than have the pipe between the boiler and the valve. Any pipe between the boiler and the valve should be as large as the area

of valve, or, better, one size larger. It should be straight and self-draining back to the boiler.

Fusible Plugs are frequently used and are required by law in boilers of sea-going vessels. They are inserted in the highest part of the heating surface and, being composed of an alloy having a low fusing point, melt as soon as they become exposed by lack of water. The blast of escaping steam gives the alarm.

The plug is generally made of composition, with a fusible metal centre about $\frac{1}{4}$ -inch in diameter. Instead of an alloy, Banca tin is a reliable metal. Tin is not liable to change its melting-point, which is about 443 degrees F. Fusible alloys do not always work, especially when old, as the melting-point of some alloys appears to change with age. Plugs, therefore, should be occasionally renewed.

The plugs should always be screwed into the boiler from the steam or pressure side, and the hole containing the alloy should be made tapering to prevent the metal being blown out. The plug should stand an inch or more above the sheet into which it is fastened, in order that it may not become covered with scale.

Various forms are in use (see Fig. 118).

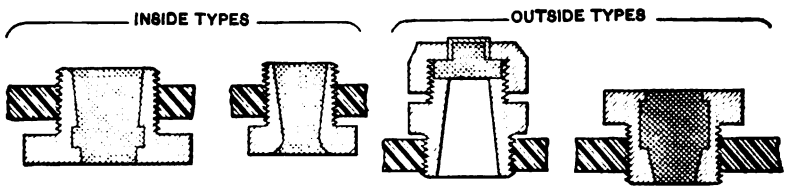


FIG. 118.—Various Forms of Fusible Plugs.

Steam-gauge. Every boiler should have connected to it a clear-faced steam-pressure gauge. The face should not be less than 6 inches in diameter, and as much more as its height above the floor and the darkness of the boiler-room may require.

Pressure gauges are made with a "Bourdon" tube, a bent tube having an elliptical section, which tends to straighten out as the pressure is applied. The extent of the straightening is magnified by a system of levers connecting the free end of the tube with the dial-pointer. The tubes are usually of brazed brass, but are also made solid-drawn, which is considered better. Some gauges are fitted with two tubes to prevent vibration of the magnifying movement, which causes rapid wear. Such double-spring gauges are useful in locomotives, fire-engines and portable boilers.

The instrument should be located at the water-line to obtain a correct reading. If located above or below the water-line, a correction must be made for accurate reading, as in testing. This correction is the weight of the water-column which hangs on the gauge according to its connection.

The connecting pipe should lead direct into the steam space and not close to any steam-pipe carrying a flow of steam, lest the current effect the pressure. If there is a superheater or steam-drum, as in Figs. 26 and 27, the steam connection should be made to the boiler and not to the superheater, as the pressure in the latter may fluctuate with the strokes of the engine. The connecting pipe to the gauge should be bent to form a trap for water, so that the hot steam may not enter the instrument. If there is danger of freezing when the boiler is not in use, a straight pipe self-draining back to the boiler can be used with a siphon-cock made for the purpose.

Continuous-recording gauges are made which record the pressures on a chart. These gauges are very convenient in some commercial works continuously operated.

Water-gauge. Every boiler should have a glass column to show the water-level, with the ends connected to the water and steam spaces. These connections should be made direct to the boiler at points as far as possible from places where strong currents may exist, as currents or fluctuations in pressure will cause the water to vibrate in the column. As a glass column is apt to break, each connection should be fitted with a valve, and the best valves are those which have an automatic closing device. The column should be so set in the fitting that steam can be blown through for cleaning. Care should be taken to see that the connecting pipes are clear, as scale or dirt in the water-pipe may close off the connection without warning. Reflex water-gauges are stronger and indicate more distinctly than plain glass gauge-tubes. They are made with grooved facets on the water side of a glass, which is so set in a metal fitting that the part filled with water appears black and that with steam shines with a silvery lustre.

Internal floats connected to a dial, like a steam gauge, cannot be recommended for general use, as the moving parts are all concealed inside the boiler. Fig. 119 shows the usual water-gauge and try-cock column; and Fig. 120 shows glass-protector guards.

Try-cocks are used to test for water-level. There are usually three, one at the high-water line, one at low-water line and one

midway between the others. When only two cocks are used, place them about $5\frac{1}{2}$ inches apart, and when three cocks about 4 inches.

These cocks are best placed directly on the boiler head or shell, but are often arranged on the glass water-column fixture. This latter plan is open to the objection that any accident to or

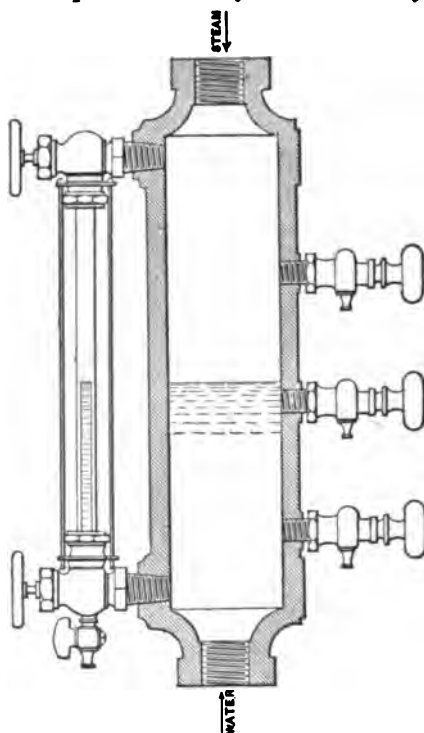


FIG. 119.—Water-gauge and Try-cock Column.

stoppage of piping renders both means useless for showing the water-level, and the boiler would have to be shut down. A drain cup and pipe can be arranged to carry off the drip from the cocks, which should not be allowed to fall on the boiler shell or head.

Some engineers place more reliance on the try-cocks than on the glass column, while others prefer two separate columns to each boiler and no cocks.

Water-alarms. These devices are used to automatically give warning when the water-level is dangerously low or high, and there are a number of makes on the market.

In general, internal floats cannot be approved, as they are out of sight and difficult to inspect and keep in repair. Internal floats are frequently used with Cornish and Lancashire boilers, especially in foreign practice. Such an apparatus is shown in Fig. 17.

A good form of water-alarm is that of a combined alarm and

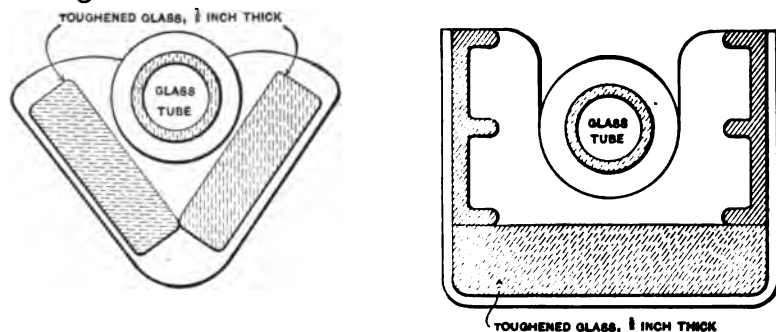


FIG. 120—Glass Protector Guards for Glass Tube of Water-column.

water-column (Fig. 121). The floats are placed inside of a cylindrical casting, which is connected at the top with the steam space and at the bottom with the water space of the boiler. These floats act on a small whistle, one opening it for low water and one for high water. The objection is the complication and liability to get out of order, due to scale and dirt, as well as the general neglect of the attendant who is prone to rely on the instrument. If such a device be adopted, the try-cocks should be on the boiler, and not on the fixture, as then the boiler could be operated by the cocks while the column fixture is shut off for repair or cleaning.

Every engineer must rely on his own experience whether such automatic "safety" appliances are a benefit, and whether their adoption more than offsets their objections.

Manholes and Handholes. Every boiler should be equipped with both manholes and handholes so located as to facilitate inspection and cleaning of all parts. In boilers that are complicated by braces, flues and other obstructions, preventing the entrance of a man, handholes should be so arranged that by placing a light through one the remaining parts can be seen through another.

These openings are generally elliptical for the convenient removal of the cover, which is placed on the inside of the boiler in order that the steam pressure will assist in holding it against the

seat. They may be any shape, however, so as to fit in between flues and braces.

The standard size for manholes is 15 inches by 11 inches, and for handholes $4\frac{1}{2}$ inches by 3 inches. When manholes are placed in a cylindrical shell, the major axis should be in the direction of

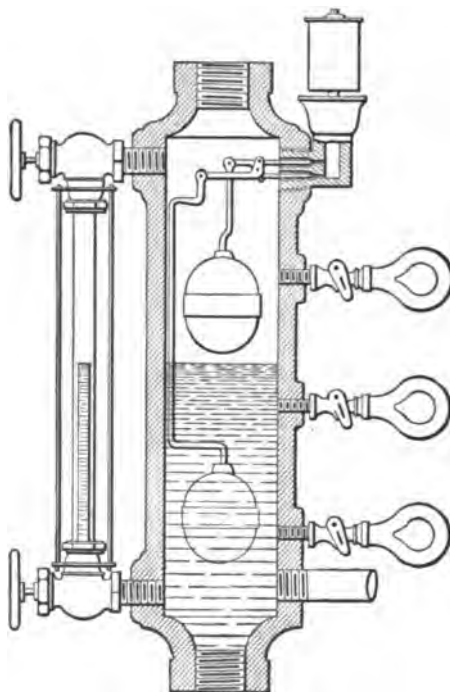


FIG. 121.—Combined High- and Low-water Alarm and Water-column.

the girth, that the least amount of shell shall be cut longitudinally, which is the direction of greatest weakness.

The opening should be reinforced to make up for the metal cut out. It is impossible to calculate the amount of reinforcement required, as the stresses are so ambiguous. The amount of compensation for the weakness caused by the hole is best determined by experience. The strengthening is usually accomplished by either riveting a flat steel ring piece around the hole, or by riveting a steel ring flanged inward, or by flanging the shell plate itself. The latter is the best method, but the most expensive, and incurs the danger of internal stresses due to the local heat for flanging,

unless the shell be subsequently annealed. The second method is generally adopted for high-pressure boilers, as it is stronger than the first or the most common plan. A great advantage in either of the flanging methods lies in the fact that the inner edge may be planed off flat when the manhole is located on a curved sheet. It is much easier to maintain a tight joint with the cover when the contact surfaces are straight than when curved. The stiffening piece may be made with an angle-ring riveted to the shell, but this method is not satisfactory, as it is difficult to make a neat appearance and obtain a close and even fit. Formerly it was quite common, especially with Cornish and Lancashire boilers, to fit a cast-iron or malleable-iron special mounting, riveted to the shell, and to place the manhole upon the top, but this method is not suitable for heavy pressures.

The covers are made of cast-iron, malleable-iron, cast-steel, or steel plate, the latter sometimes forged into a convenient corrugated section for strength.

When fastened to a special mounting, the covers are usually bolted to the flange of the mounting with bolts spaced not over seven thicknesses of flange apart. Ordinarily the covers are held in place by yokes or dogs and bolts. For large manholes there should be two yokes, and for small ones and for handholes one yoke will suffice. There is generally but one bolt to each yoke. As the cover is on the pressure side, the object of the bolt is to draw the cover up tight. The feet of the yoke should rest on the boiler-sheet or stiffening-ring, with its axis at right angles to the major axis of the opening.

The joint between the cover and the boiler should be smooth and a true fit. It is kept tight by a gasket of hard rubber, thin asbestos, or corrugated copper, or by a copper wire or fine lead pipe laid in a groove and squeezed tight by the pressure of the bolts. Gaskets cut from sheets of packing or rubber are the simplest. Corrugated-copper gaskets make an excellent joint and can be bought ready made of standard sizes.

The accompanying Figs.* 122 and 123 illustrate typical manholes, the details of which can be varied to suit special requirements.

* See Marine Engineering, June, 1899.

Grates. The grate may be of either the stationary, shaking, or mechanical-stoking class. Mechanical-stoking grates will be discussed in Chapter X.

The grate should be so formed as to allow sufficient air to pass

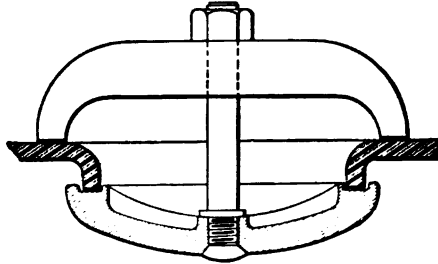


FIG. 122.—Manhole and Cover for a Flat Sheet.

through it, and the openings should not be too large or the coal will fall through unburned.

Provision for allowing air to enter above the grate should always be provided, more especially with bituminous coals and wood. For anthracite, the area of such openings should be about $\frac{1}{8}$ of the grate surface, and for very bituminous coals about $\frac{1}{16}$ of grate

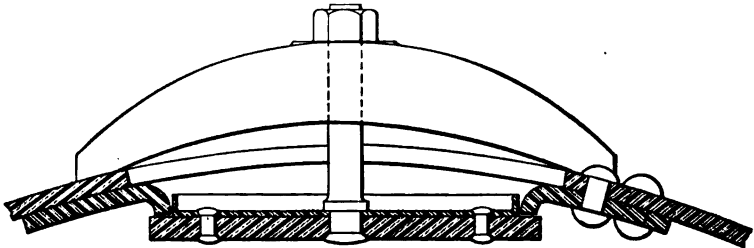


FIG. 123.—Manhole and Cover for a Cylindrical Shell.

surface. Intermediate coals should have a proportionate ratio. Such liberal proportions are seldom found in practice. Some area can be provided in the fire-doors, some over the doors as in Fig. 12, and some in a split bridge, or in a passage through the bridge wall as in Figs. 124 and 125, but all must be equipped with doors and dampers so as to shut off or regulate the air-supply.

The distance from the grate to under side of shell in externally fired boilers should be, with anthracite coal, about 24 inches for grates 4 feet long, and be increased in proportion. For bituminous,

non-caking coals, this distance should be increased to from 30 inches to 36 inches; and for fatty coals from 40 inches to 48 inches, while even greater heights have been found beneficial.

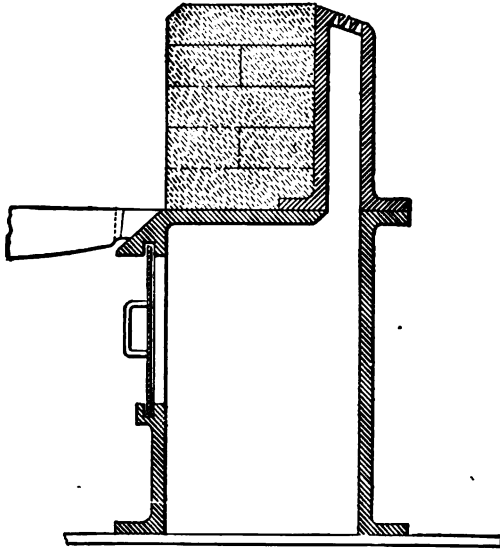


FIG. 124.—Split Bridge with Passage to Admit Air.

The height of grate above the fire-room floor should be from 18 inches to 26 inches, 24 inches being about the average.

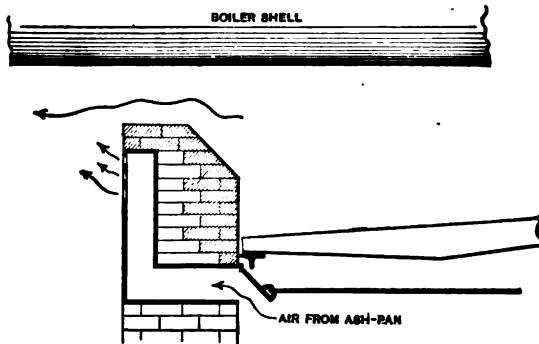


FIG. 125.—Split Bridge with Passage to Admit Air.

In internally fired furnace-flues these distances cannot be realized, but the larger the flue the better, especially with soft

coals, and with small flues use a short grate. Grates should not be longer than twice the flue diameter in order to accomplish the best results.

Stationary grates are generally made of cast-iron. Cast-steel makes an excellent and durable grate, but the castings are somewhat difficult to make, as they are liable to warp when cast.

Grate-bars are usually cast in lengths of 24 inches, 30 inches and 36 inches, consisting of two or three bars to a set (Fig. 126).

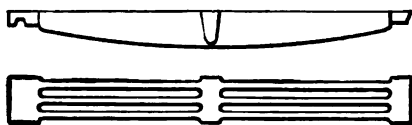


FIG. 126.—Cast-iron Grate-bar.

The bars are made tapering downward, to enable the ashes to drop clear. The upper part is best made parallel for about $\frac{3}{4}$ inch, so that the upper surface may burn away to that extent before the openings are increased in width. The bars are made deep that the entering air may be heated, thus keeping the bars cooler and preventing them from burning and twisting. The bars are usually $\frac{3}{4}$ -inch wide at the top, tapering to $\frac{3}{8}$ -inch at the bottom, and are from 3 inches to 4 inches deep at the centre. As depth is advantageous, the above amount could be increased when there is plenty of height in the ash-pit. The bars are cast with distance-pieces or lugs to keep the proper spacing and prevent warping. The tops of the bars are often grooved for use with bituminous coals. These grooves admit air to all parts of the fire, and tend to prevent clinkers from attaching to the bars on account of the ashes that collect in them.

The aggregate area of opening between the bars should be about fifty per cent of that of the grate, but not less than forty per cent nor in excess of sixty per cent. The fatty or gaseous coals require a larger free area than hard coals. For very fine coals and for some grades of cheap coals and lignites, a perforated plate can be used with advantage.

The width of opening depends on the quality and size of coals, hard coals requiring narrower openings than soft coals. Too wide a spacing of the bars cannot be used with the bituminous caking coals, as they will form solid clinkers difficult to break out.

In general, the narrower the bars and spaces the better, but on account of excessive warping very thin bars cannot be used. The average proportions are about as given in Table XX.

TABLE XX
AIR SPACES AND THICKNESS OF GRATE-BARS

Size and Kind of Coal.	Width of Air Spaces.	Thickness of Grate-bars.
Screenings.....	$\frac{1}{2}$ inch	$\frac{1}{2}$ inch
Anthracite, average.....	"	"
" buckwheat.....	"	"
" pea or nut.....	"	"
" stove.....	"	"
" egg.....	"	"
" broken.....	"	"
" lump.....	1	"
Bituminous, average.....	"	"
Wood, slabs.....	"	"
" sawdust.....	$\frac{1}{2}$ to	"
" shavings.....	$\frac{1}{2}$ to	"

Grates often are set horizontally, but long grates are best placed sloping toward the rear to facilitate firing. One inch per foot is a good allowance.

The front of the grate, when designed for bituminous coal, is often made solid. This piece is called the "dead plate." The fresh charge of coal is thrown on the dead plate and allowed to coke until all the hydrocarbons have been volatilized and burned as they pass over the incandescent coal in the rear. The charge is pushed back on the open grate at the time of next firing. The dead plate is not regarded so favorably as formerly, due to the high rates of combustion now practised. In order to coke all the coal at rapid rates of combustion it would necessitate a great width of dead plate and consequently an inconvenient length of grate.

Grates should be so supported that one end is fixed and the other free for expansion. With brick-set boilers, the bearers can be built into the brickwork. With internally fired boilers, the proper placing of bearers is more difficult. It is bad practice to fasten clips by tap-bolts to a fire-box or flue, as these bolts are apt to leak and corrode. For upright vertical boilers a cast-iron plate suitably shaped and placed under the water-leg, as in Figs. 13, 14, 15 and 15a, is much better. In furnace-flues, a bearer with wide tee-shaped ends made to fit the sides of flue will answer, as the fric-

tion will lend all the support necessary. In corrugated furnaces, a half-round steel bent to fit into a corrugation, as in Fig. 127,

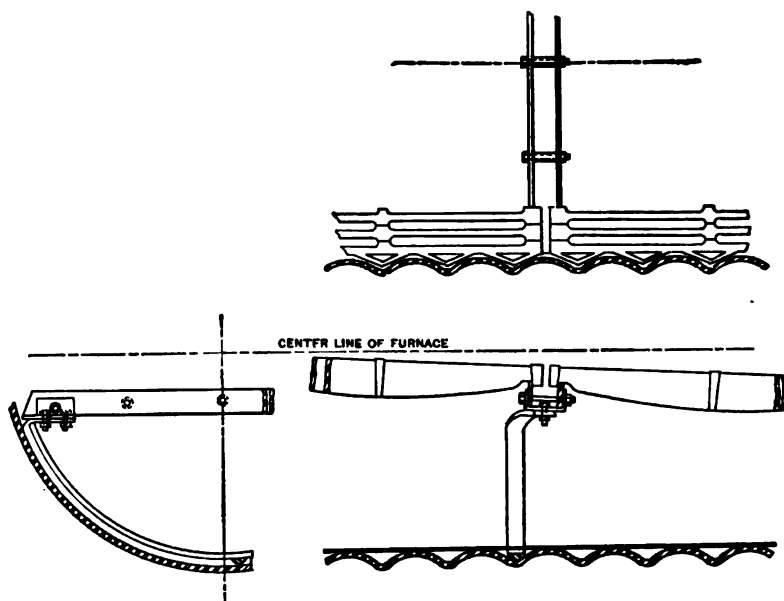


FIG. 127.—Grate Bearer for Corrugated Furnace.

will be sufficient. When properly made, there is no danger of the grate falling with either of these latter methods, as sufficient longitudinal support is received from the front and bridge wall. In corrugated furnaces the outside bars must be specially made to fit close into the corrugations. If an ash-pan of curved iron plate is used in the furnace-flues, the clips for bearers can be bolted to the pan without objection.

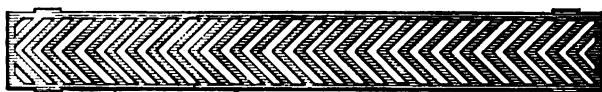


FIG. 128.—Plan of Herring-bone Grate-bar.

Instead of straight bars, the herring-bone pattern is frequently adopted (Fig. 128). The angular cross-bars are durable, and give freedom for expansion combined with great stiffness in the wide casting. They are not so easily sliced as straight bars, and are, therefore, not so convenient with coals that clinker badly.

Shaking-grates have an advantage in affording means to clean out the ashes and dislodge clinkers without opening the fire door. This is the real base of the claims for increase in economy. They also save considerable manual labor and are therefore much liked. Those forms are to be preferred which have the least complications, and which break up the fire with the least disturbance of the bed of coals.

There are a number of varieties of sectional shaking-grates on the market, and some of them are made self-dumping (Fig. 129).

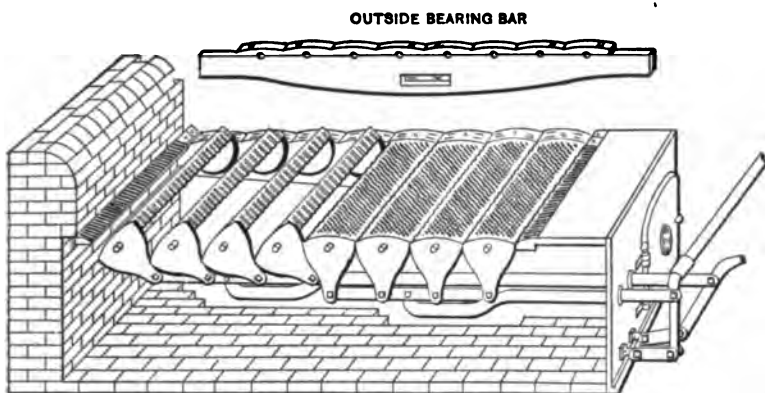


FIG. 129.—Shaking-grate.

The Ashcroft grate consists of steel bars of triangular section that can be turned alternately by a key, fitting the ends which project through the front (Fig. 130). These bars are supported

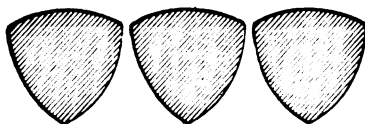


FIG. 130.—Section of Ashcroft Grate-bars.

by bearers spaced about 18 inches apart, made of a steel strip set on edge, with half circles cut out to fit and guide the grate-bars. The bars are liable to warp and twist; but as they can be straightened, they should be made easily removable. This makes a very light form of grate.

For burning sawdust, tanbark and similar fuels, a grate-bar of the form shown in Fig. 131 is often used.

The **Down-draft Grate** is a special design, so arranged that the draft is downward through the bed of coals (Fig. 132). Below the upper grate is another on which the falling particles of coal are



FIG. 131.—Grate for Burning Sawdust or Tan-bark.

caught and burned. Beneath the lower grate is the ash-pit. The upper grate is made hollow and has a water circulation so that it may be kept cool and prevented from burning. Incidentally

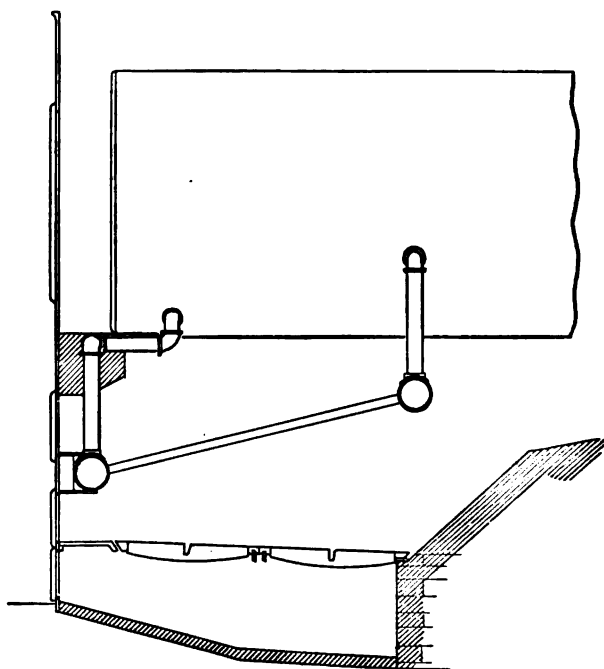


FIG. 132.—Down-draft Grate.

this increases the heating surface. Tests with certain grades of fuel have shown this downward draft very effective, and owing to the control that can be placed on the air-supply, it is promising of economic results.

After passing down through the grate, the air and products of combustion enter a very hot combustion-chamber. Into this com-

bustion-chamber the balance of air requisite for combustion can be readily admitted. A strong draft is required, as the upper grate generally is relatively small in area and the rate of combustion correspondingly high. The coal should be worked by the fireman, in order to keep the draft clear, which involves extra labor. Therefore the grate always will be more or less clean, which will tend to reduce the capacity of the furnace to respond to a sudden demand, as there will be no ash to shake out in order to freshen the fire.

For a given boiler, the capacity of a down-draft grate is less than that of an up-draft grate of the usual size, but more than an up-draft grate of equal area. It appears from experiments made at St. Louis (see Bryan on Down-draft Furnaces, *Trans. Am. Soc. M. E.*, Vol. XVI, 1895) that less surplus of air is required with this form of grate than with one of the ordinary type. That reducing the grate area from the usual proportions showed an increase of economy but caused a considerable reduction in the boiler's capacity for over-work.

Ash-pit. The ash-pit should be sufficiently high to easily admit the air required for combustion. Small ash-pit doors are a too frequent fault. In fact the doors themselves are sometimes sources of danger, since careless firemen will use them to check the draft, instead of the damper in the uptake, and thus burn out the grates.

The height of the ash-pit may be as much as desired and convenient, but, if possible, should never be less than 24 inches, except with very small boilers. As the height of grate above the fire-room floor is fixed by convenience for charging coal, the ash-pit may be sunk below the floor in order to secure increased height. The entrance should then be inclined as in Figs. 11 and 12.

Some engineers inject steam into the ash-pit and others use a tight pan with water, both with the object of assisting combustion. There is really no advantage in economy beyond the partial prevention of clinkering.

In furnace-flues an ash-pan is often used, made of $\frac{1}{4}$ -inch or $\frac{3}{8}$ -inch iron plate curved to fit. They are not necessary in plain flues, but are useful in the corrugated forms. The only object is to provide a smooth surface for raking out the ashes. Instead of using an ash-pan, the corrugations can be filled flush with cement.

When the ashes are pulled out of the pit or the fire drawn, the refuse should not lie against the boiler front, as it will soon become corroded. For furnace-flue boilers, as the Cornish, Lancashire and Scotch, there should be an iron apron to protect the head. This apron can be easily renewed.

Fire-doors. The fire-doors are made of cast-iron, cast-steel or wrought-steel. They are seldom made less than 12 inches high by 16 inches wide. For very wide grates two small doors are preferable to one large one. The door is protected on the inside by a liner plate, which should be perforated to break up into fine streams the air entering through the door damper. These liners are made of cast-iron and should be removable (Figs. 133 and 134).

The doors can be made to automatically shut the up-take damper when they are opened. These automatic arrangements are excellent in principle, especially with forced draft, but are liable to derangement and have often proved unsatisfactory in practice.

Breeching, Uptake and Smoke-connection. Unless the chimney rests directly on the boiler, a connection must be made, and this smoke-pipe is called the breeching, uptake or smoke-connection. It always must be used with boilers set in battery, delivering the products of combustion into a common stack.

The connection when under the fire-room floor may be a brick-lined conduit. When overhead, it is made of steel, either square or round in section. The former shape is the stronger and the latter the cheaper.

The connection should increase in area as additional boilers are connected, otherwise the boiler nearest the stack will have the strongest draft. The connection should be free from all sharp bends, and branches should not be made directly opposite one another.

Draft Regulators. The draft can be controlled by a regulator so that a constant steam pressure may be maintained. They do not relieve care of the fire, but simply open or close the damper as the steam pressure falls or rises. They are economical and very useful, especially in plants where the demand for steam fluctuates rapidly.

In the most common form, the boiler steam presses on a diaphragm, which is connected to a lever that controls a small water valve. If the steam pressure falls, the water valve is opened, permitting water to escape from a hydraulic cylinder, thus lower-

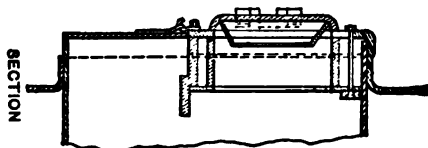
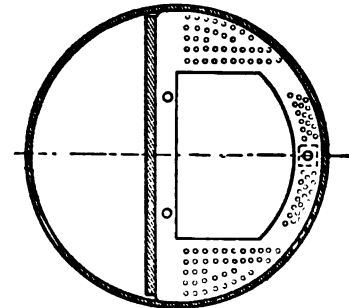
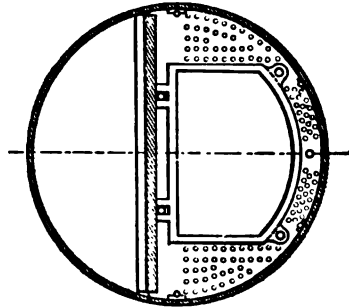
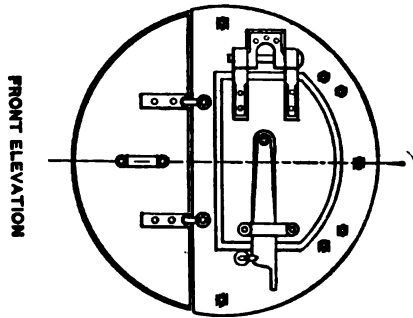


FIG. 133.—Fire-door for Furnace flue.

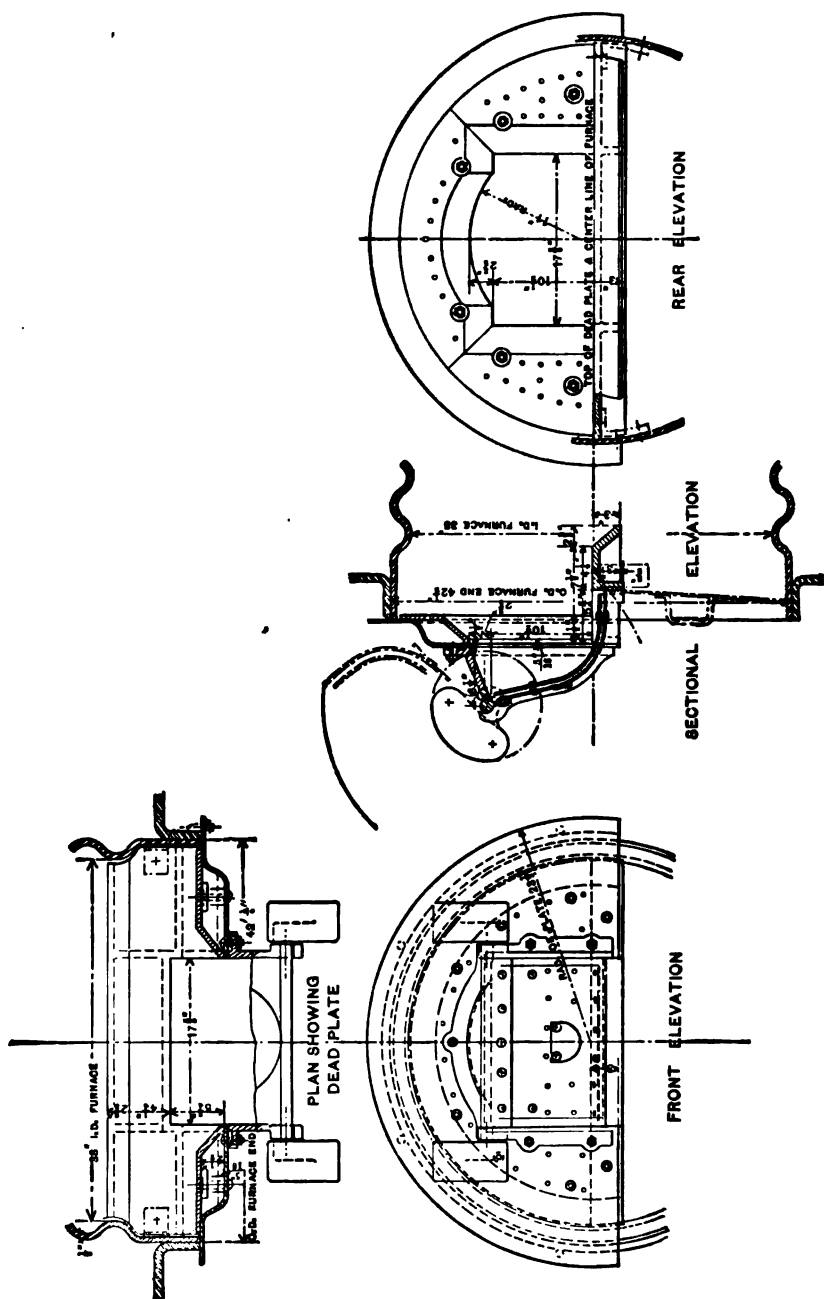


FIG. 134.—Morison Furnace-door and Furnace-front.

ing the plunger and opening the damper. If the steam pressure rises, water is admitted to the cylinder, which raises the plunger and closes the damper. When properly adjusted, they work in a very satisfactory manner.

Steam-traps. In a system of steam-piping it is often convenient to lead the drip-pipes which carry off the condensed steam to a trap. This trap, while preventing the escape of steam, will discharge the water and keep the system drained.

When the trap discharges freely into a cistern, a hot-well or a sewer, it is called a **Discharge Trap**; and when it discharges back into a boiler under pressure, a **Return Trap**.

There is a great variety of traps on the market, but the best forms are those which are least complicated and the quickest and most readily examined.

A Kieley discharge trap is shown in Fig. 135. This is of the

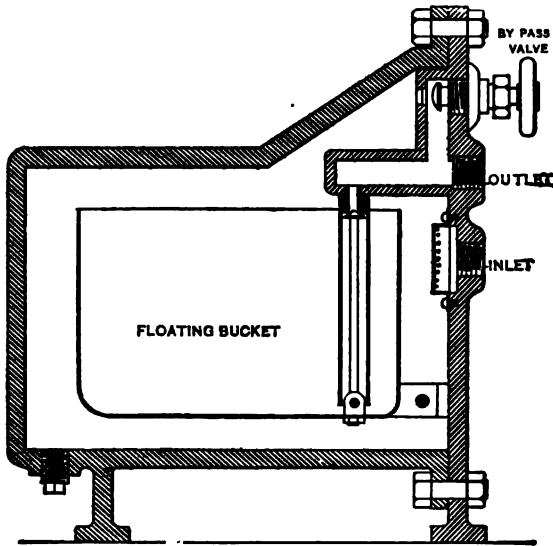


FIG. 135.—Kieley Discharge Trap.

bucket variety. The water floats the bucket and closes the outlet. When the water rises and overflows into the bucket, it sinks and the pressure discharges the contents until the bucket again floats and closes the outlet. The casing is so made that it can be removed and expose all the parts without disconnecting them.

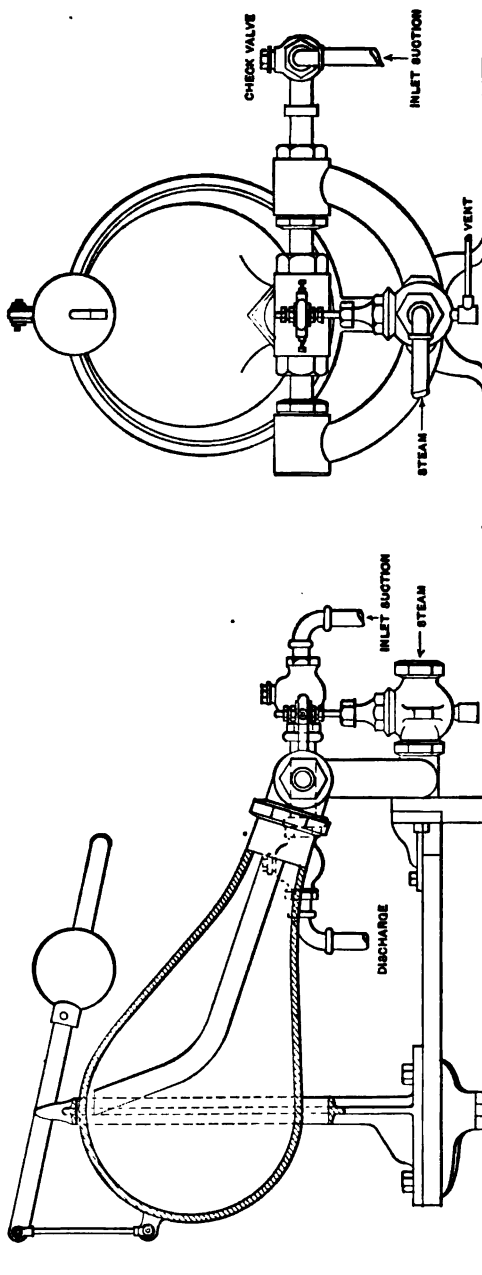


FIG. 136.—Bundy Return Trap. Must be located above the water-line of boiler.

Return traps are located above the boiler, about 18 inches or more, according to the steam pressure, so that the trap may empty by gravity. The various drips are led to a manifold, which is connected to the trap. A Bundy return trap is shown in Fig. 136. The water enters through one of the trunnions until the weight of the pear-shaped vessel overcomes the balance-weight and falls,

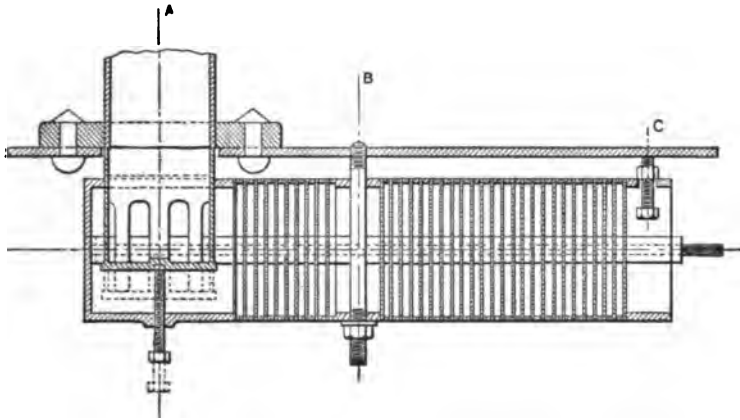


FIG. 137.—“Potter” Mesh Separator—Longitudinal Section.

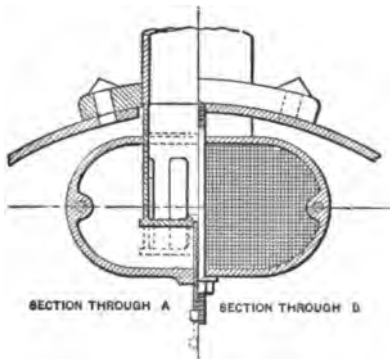


FIG. 137a.—Cross-section of Fig. 137.

thus opening the equalizing valve which admits steam from the boiler through the other trunnion. The equalizing of the pressure in the trap and boiler permits the water to flow into the boiler. Check-valves are inserted in the pipe connections to prevent the water flowing the reverse way.

Separators are devices placed on a steam-pipe to remove the priming or entrained moisture carried along with the steam. Moisture affects the steam by increasing its density and its heat conductivity. It should be avoided, therefore, as it augments serious losses in the piping and at the engine through losses of heat by radiation and through changes of temperature in the cylinder walls. Moisture in quantity may cause damage by water-hammer, which frequently is of a serious nature. While not essential to the well working of a boiler, they are much used and a valuable accessory.

Separators are designed on two principles, either by inserting a plate or plates against which the current of steam strikes, permitting the moisture to flow down and drip off, while the gas passes around the obstruction, also depositing more water by the change in direction; or by inserting a spiral passage for the steam so that the moisture is thrown out by centrifugal action.

In Fig. 137 is shown a Potter mesh separator, consisting of a series of plates; and in Fig. 138, a Stratton separator, operated on the centrifugal principle

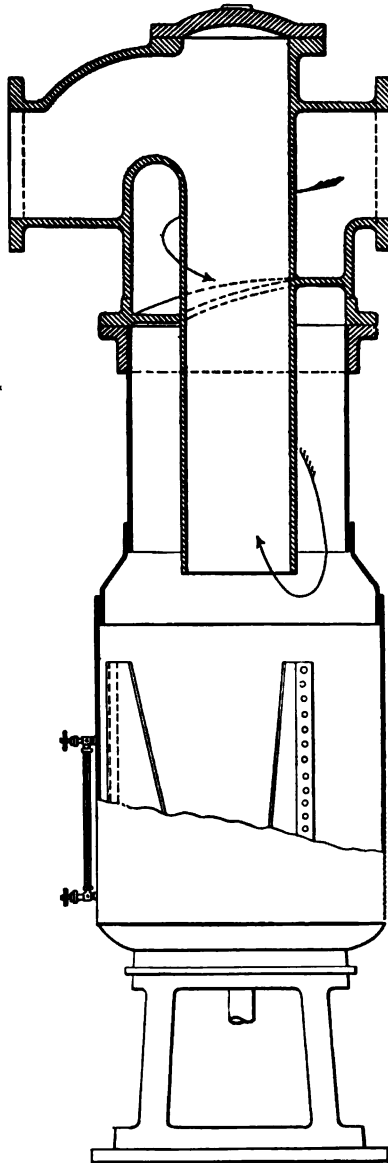


FIG. 138.—Stratton Steam-separator.

The separator should be located as close to the engine as the

convenience of the general arrangement will permit. The amount of water collected in the separator can be seen in the glass water-column attached, and can be blown out as required either into the hot-well, condenser or sewer drain. The water may be automatically removed and discharged by a trap, which is the more satisfactory plan.

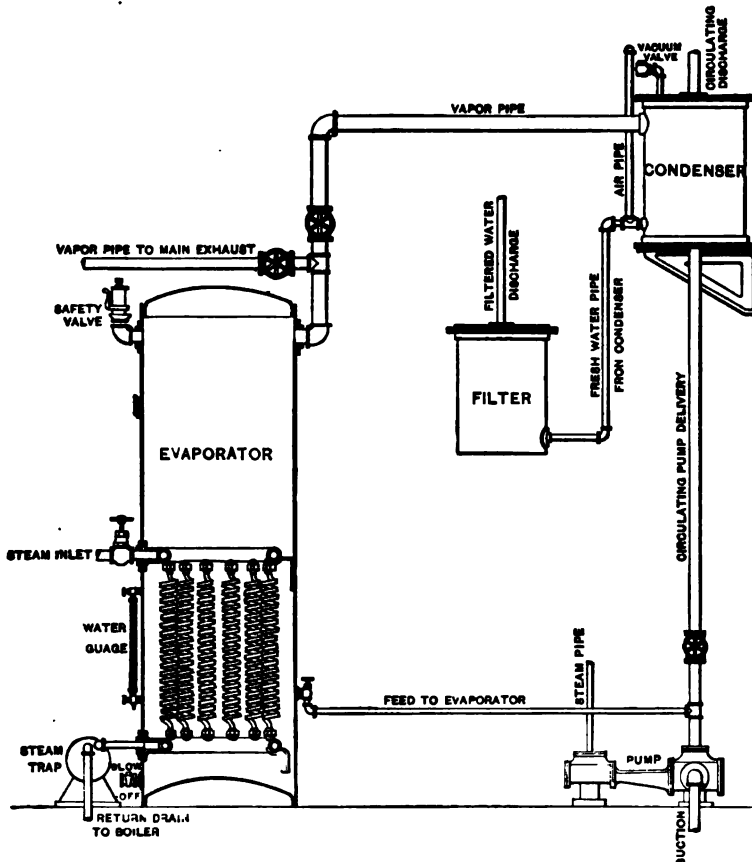


FIG. 139.—Salt-water Evaporator.

Evaporators are used to replenish the supply of fresh water for boiler-feeding when salt water alone is available. They are therefore most used on shipboard and avoid the necessity of carrying large storage-tanks, thus reducing weight and increasing cargo capacity.

Salt water is very objectionable when steam is used at pressures exceeding 100 pounds per square inch, and always should be avoided in water-tubular or sectional boilers.

The evaporator is in reality a special form of boiler, with heat supplied by steam from the main boiler, and arranged to conveniently blow out the salt as it deposits.

In Fig. 139 is shown a form of evaporator as made by the James Reilly Repair and Supply Company, of New York. It consists of a steel shell containing a copper piping system through which steam passes. The condensation is removed by a trap, and discharged into the hot-well, condenser or boiler. An auxiliary pump forces water into the evaporator as required, through a by-pass pipe, while the main supply circulates through a special condenser for condensing the vapor. The heat of the steam evaporates the salt water, and the steam or vapor passes into the special condenser, from which the fresh water may be drawn either to the hot-well or the condenser. It also may be made to pass through a filter for furnishing drinking-water. If desired the steam from the salt water can be passed to the main condenser of the engine, or to the low-pressure valve-chest of a triple-expansion engine and be made to work in the low-pressure cylinder.

There are many forms of evaporators on the market, but, like all boiler accessories, they should be simple and easily cleaned and repaired.

CHAPTER X

MECHANICAL STOKERS

Classes, Over-feed and Under-feed. Advantages. Disadvantages. Results Obtained by Use.

Mechanical stokers operate on two principles, those which "over-feed," or spread the fresh coal on top of the fuel bed, and those which "under-feed," or push the coal forward beneath the grate until it overflows out on the grate (Figs. 140, 141, and 142). In the latter operation the coal is coked as it nears the fire in

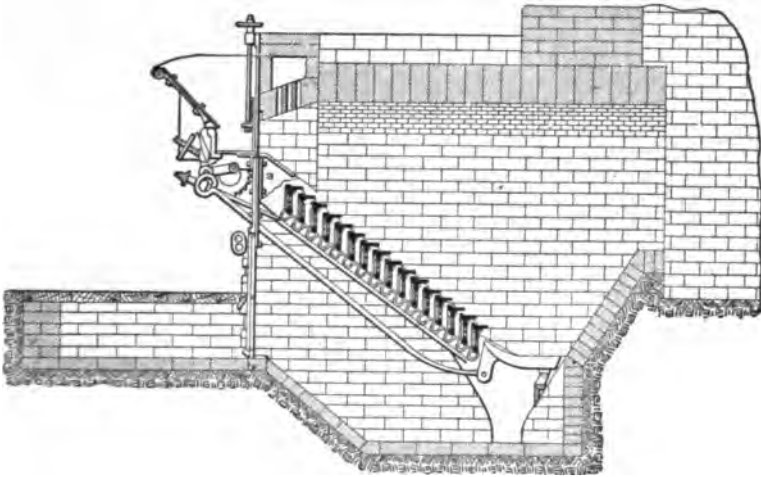


FIG. 140.—The Roney Mechanical Stoker.

its forward and upward course. Stokers operating with a revolving endless grate, such as the Coxe Mechanical Stoking Furnace, are a special form of the over-feed class.

The fuel is fed through a hopper and the rate of feeding is controlled by a motor. Any kind of coal, wood blocks, shavings, etc., can be used.

The claims made in favor of mechanical stokers are: A steady and uniform supply of fuel; diminished danger of burning holes in the fire, thus letting air pass in large quantities at places where the least amount is required; smoke prevention due to continuous firing in small quantities and controlled air-supply; reduction of labor; and the opportunity to use coal-conveying machinery to best advantage.

The objections generally cited are: Complications; lack of

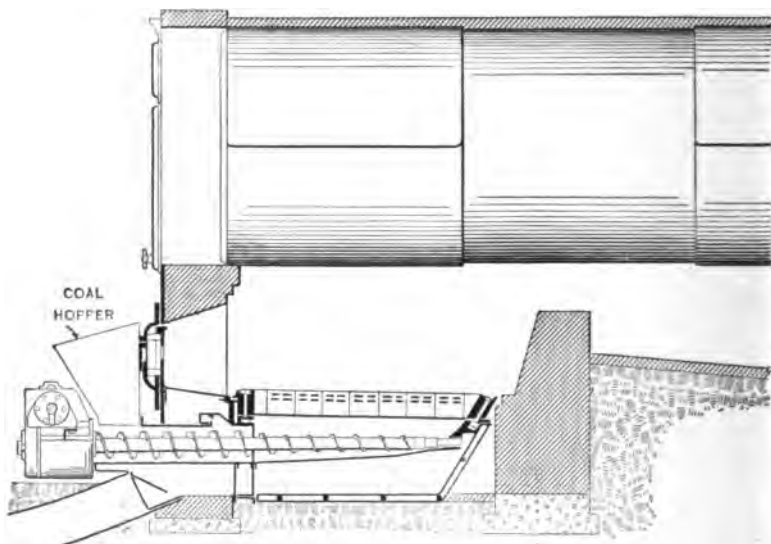


FIG. 141.—The American Mechanical Stoker.

durability; lack of reliability to feed the fuel evenly over the fire grate; and lack of facility to force the fire in response to sudden demands.

With the best forms of stokers, the advantages claimed are more or less attainable, while the objections are not always realized. A cheap stoker, however, is a poor investment. Only the best should be adopted, for if there is to be a saving, such saving will repay the difference in cost. Stokers should be operated in strict accordance with the principles of their design. In short, the stoker is intended to do a certain work, and the attendant should assist, not force, it in the performance of its duty. The best stokers are those which are least complicated, have the fewest parts to

get out of order and have the details properly worked out in accordance with good mechanical principles.

Considerable information can be obtained from the makers' catalogues, but such publications must be read with the usual care.

The Steam Users' Association, of Boston, has published some valuable statistics, from which the following has been taken: *

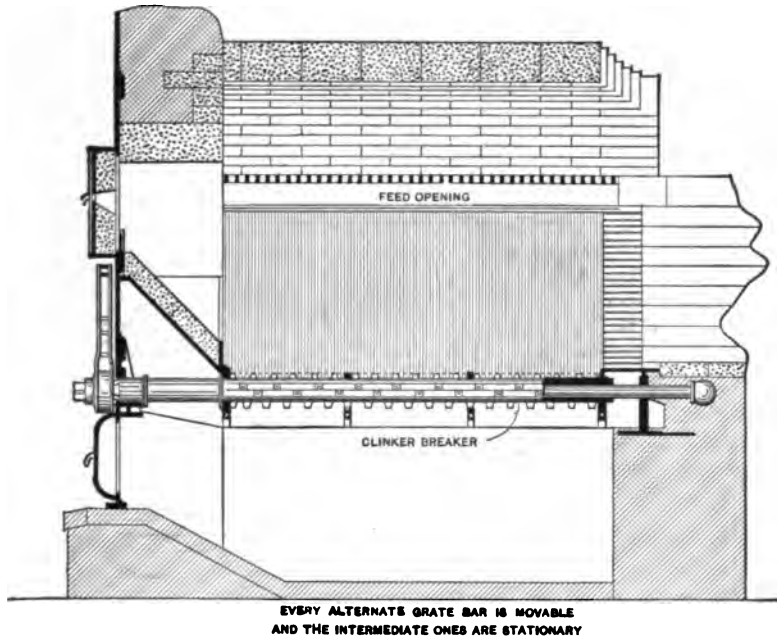


FIG. 142.—The Murphy Mechanical Stoker.

"Stokers may save a slight amount of coal. In calculating this from an evaporation test the amount of coal used by the stoker engine and steam blast must be allowed for. This may reach 11 per cent.

"Stokers save labor in large plants, provided coal-handling machinery is also installed.

"Stokers save 30 per cent to 40 per cent of labor in very large plants (using over 200 tons of coal per week) 20 per cent to 30 per cent in medium-sized plants (50 to 150 tons per week), and save no labor in small plants.

* Steam Users' Assn. Circulars Nos. 7 and 9. Report by R. S. Hale.

"Stokers save smoke in all plants.

"Stokers cut down the capacity, but not to any great extent, and this may be made up by extra draft.

"Stokers, on an average, reply to a sudden call for steam as quickly as hand-firing.

"The repairs on stokers are not excessive when the fire-room force has become experienced in their use, but may be very heavy with inexperienced firemen. It is sometimes claimed that the use of stokers makes the boiler repairs less than with hand-firing.

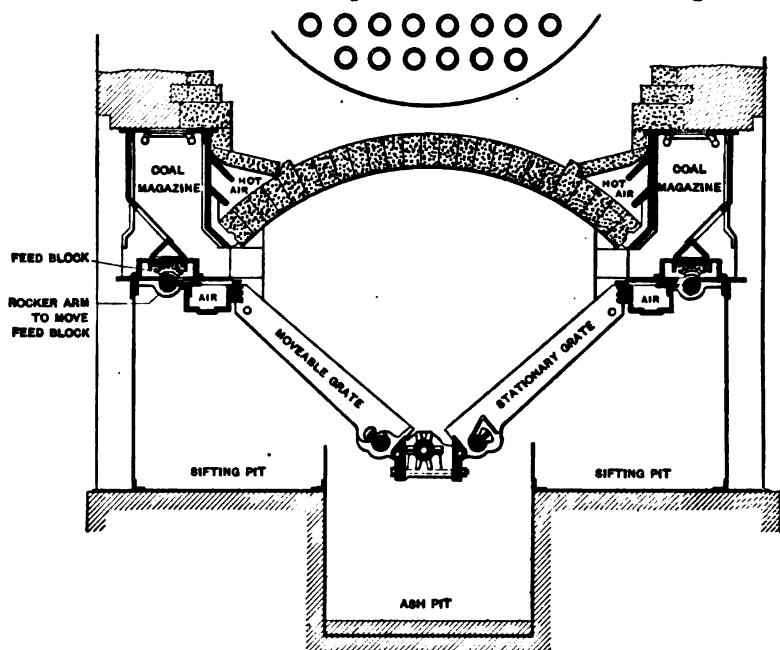


FIG. 142a.—Section of Fig. 142.

"In a large plant stokers will be advisable if they make possible the use of a cheaper fuel than can be fired by hand. But it should be ascertained that the cheaper fuel cannot be used without the stoker, since many patent devices claim and obtain credit for savings due to other causes. In such cases there will be a saving in cost of fuel and a considerable saving in labor and smoke, while it seems unlikely, judging from the data collected, that the increased interest, repairs, depreciation and power used in running and in steam blast will be enough to overcome these gains.

"If no gain can be made by using a cheaper fuel, still stokers may be advisable in large plants burning a poor grade of soft coal. In such cases the saving in coal will not be so great as when the stokers cause a saving by using a cheaper fuel, but there is probably some slight saving in the coal, and at any rate a saving in labor and smoke. The interest, repairs, depreciation and steam for power and blast may or may not balance these savings.

"In small plants stokers will be seldom advisable unless the saving in cost of fuel will be quite large, or unless the smoke nuisance is serious. In small plants the labor saving is small or even less than nothing, while the expenses are no less in proportion than in large plants.

"Mechanical stoking differs from hand-firing in that the firing is continuous, and generally in that the grate-bars are in continual motion. These grates are generally smaller than hand-fired grates.

"The smaller grate of necessity reduces the capacity. The motion of the grate-bars is continuously disturbing the ash films and so increasing the rate of combustion per square foot of grate. This largely makes up for the small size of the grate, when compared on a long test, but the capacity to meet a sudden call is less than in hand-firing, when the hand-firing is clean. The stoker, however, never gets as dirty as a hand-fired grate will after a few hours, and the clean stoker fire responds more quickly than the dirty hand fire.

"The continuous firing has this effect: There is a certain supply of air per pound of coal which gives the best results, too much or too little resulting in a loss. If the firing be continuous the air-supply can be adjusted to fit the firing. If the firing be intermittent, as in hand-firing, then the air-supply is first too small, then too large, and a loss results. The more intermittent the firing is, the greater is the average variation from the proper air-supply, and the greater is the loss. The air-supply per pound of coal shows a much greater tendency to vary with soft coals than it does with hard. Therefore the softer the coal, the more saving by a stoker. The stokers are very apt to drop good coal into the ash-pit, due to the continuous motion of the grate-bars, but as it is easy to make arrangements for catching and refining this coal, this is unimportant."

CHAPTER XI

ARTIFICIAL DRAFT

Advantages. Disadvantages. Classification. Selection Depends on Local Conditions. Boiler Must be Suited to Draft. Vacuum and Plenum Systems Compared. Economy. Intensity. Jet in the Stack. Jet under the Grate. Fans. Power Required. Closed Ash-pit. Closed Fire-room. Induced Draft.

Artificial draft is steadily growing in favor with steam-users. Through its aid the steaming capacity of a boiler plant can be increased; the necessity for a tall stack or chimney dispensed with; a high economy obtained (under proper conditions) due to increased furnace temperature, produced by a more rapid rate of combustion and a reduced amount of air-supply in proportion to the fuel consumed; low grades of coal or cheap fuels burned; and positive control of the furnace maintained so as to suit changes in operation or weather. The chief disadvantages are liability to injure the boiler due to careless manipulation; cost of operation, maintenance and repair; extra complications, and risk of derangement.

The costs for interest and maintenance of a stack necessary to produce a natural draft of equal intensity will offset in a large measure (and sometimes entirely) the cost of operating an artificial-draft system.

An artificial-draft system can be designed to consume the fuel either at a low or a high rate. In the latter case the system is commonly known as forced draft, and it was for this purpose originally intended.

Artificial drafts are best classified into "jet drafts" and "mechanical drafts." Mechanical drafts are again subdivided into "forced draft" and "induced draft." With forced draft the air is forced through the furnace by mechanical means, and with

induced draft the air is sucked through the furnace and the products of combustion are discharged into the stack by mechanical means. Considerable information on various forms of mechanical draft can be obtained from a catalogue publication entitled *Mechanical Draft*, issued by the B. F. Sturtevant Company, Boston, Mass.

In general, it is not possible to state which system of artificial draft is the better, or which should be adopted. So many considerations have to be taken into account, that each case must be worked out and settled on its merits. While a system of artificial draft has attractions, there are always surrounding conditions which have their influence on the selection of the draft problem, that can neither be overlooked nor undervalued.

Artificial drafts produce either a partial vacuum or a plenum in the furnace. It is a mooted question which system is the better, so much depending on installation considerations. However, the induced or partial vacuum systems do not appear to have so marked a tendency to burn holes in the fire and produce blow-pipe effects.

The jet drafts have not proved as economical as the mechanical drafts, while between the various forms of mechanical draft sufficiently reliable results have not been obtained to make any fair comparison. Whatever may be the difference it is certainly not great.

When artificial draft is decided upon, the boiler must be suited to the high temperatures, have sufficient heating surface to absorb the heat from the gases and have a strong and effective circulation. Some boilers will often do well with a natural draft, but leak under an artificial draft on account of the rapid changes in temperature produced when the fire-door is opened. The expanded tube ends of fire-tubular boilers are frequently thus affected, and ferrules have to be used. These ferrules may prevent a continuance of the leak, but usually do not remove the cause of the trouble.

The strength or intensity of the draft is expressed in "inches of water" or in "ounces per square inch," with both the vacuum and plenum systems. It is usually measured at the base of the stack, but sometimes in the ash-pit and in the furnace. "Inches of water" expresses the difference in height of the two columns of water in a manometer or U-shaped tube, one end being exposed

to the draft and the other to the atmosphere. "Ounces" expresses the weight of this height of water reduced to that unit per square inch.

Steam Jet in the Stack. This method is not economical, as the amount of steam expended is large. Under some conditions, as in the locomotive or fire-engine, it is necessary, since with present designs no other method has proved so commercially good.* The

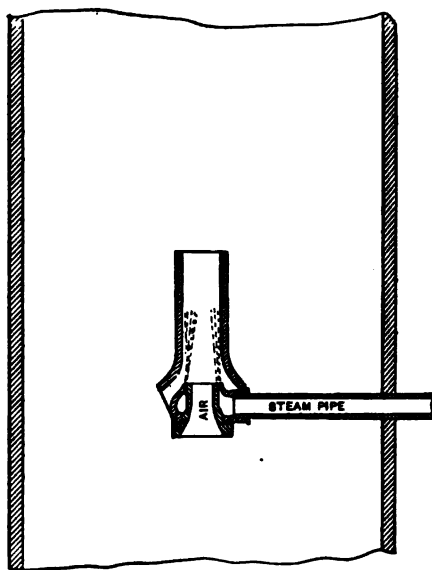


FIG. 143.—Bloomsburg Jet in Stack.

jet also is extremely useful when steam has to be raised quickly, or in providing for sudden calls for increase of steam during short periods.

The apparatus is simple, very light, easily arranged, not liable to derangement and very effective in producing a moderate draft. It consists of a jet or of a pipe with perforations, placed at the base of the stack, so that the jet or jets of steam are discharged upward, thus causing a flow of the gases in the stack independent of their temperature (Figs. 143 and 144).

The steam connection to the jet should be made direct to the

* With these engines there is available exhaust steam at high pressure, and a boiler relatively small to the size of the engine may be used to advantage

boiler and not to any steam-pipe. The size of pipe required depends on local conditions, but ordinarily a 1-inch or a $1\frac{1}{2}$ -inch pipe is all that is necessary. The pipe should be led direct to the jet and be fitted with a stop-valve easily controlled from the fire-room floor.

A jet, formed from 1-inch pipe, can consist of a cross, of a

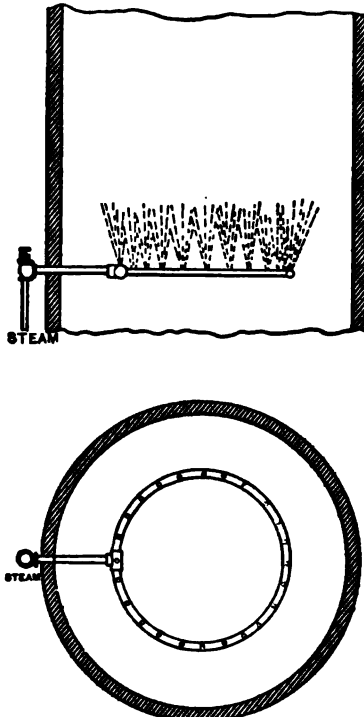


FIG. 144.—Ring Jet in Stack.

ring or of a series of rings arranged conically, all having holes (or slots) about $\frac{1}{8}$ - or $\frac{1}{4}$ -inch in diameter on the upper side only. A number of small holes well distributed will be more effective than one large hole of equal area. When not in use these pipes do not offer any material resistance to the ordinary draft (Fig. 144).

The jet may be produced by the exhaust steam from the engine, as in locomotives. In such cases of exceptionally strong blasts, the vacuum may amount to from 4 to 8 inches of water.

Steam Jet under the Grate. A jet under the grate can be arranged to produce a powerful draft by forming an air-pressure in the ash-pit. It has a stronger tendency to burn holes in the fuel than the jet in the stack. The entering steam heats the air which it draws in by inspiration through a suitable opening surrounding the jet, and also forms water-gas with the incandescent

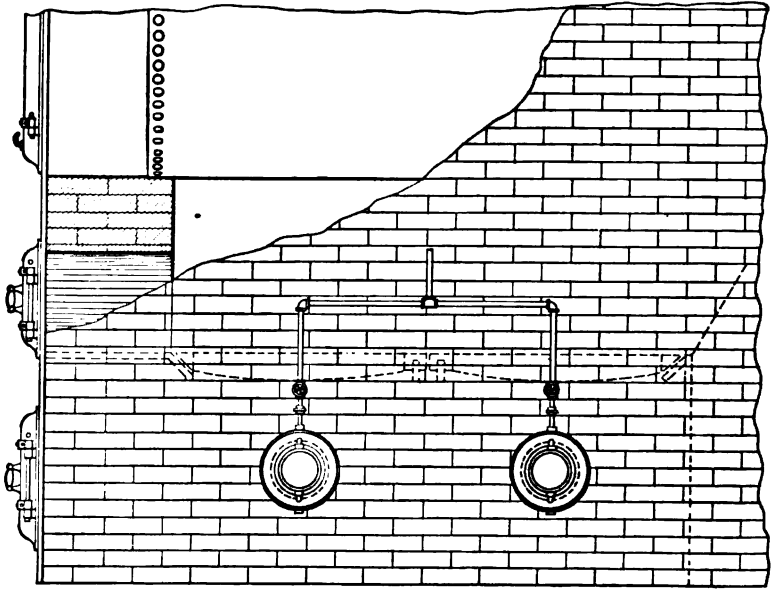


FIG. 145.—Beggs' Argand Steam-blower.

fuel. The steam also tends to prevent the formation of clinkers and thus assists combustion.

This system of forcing the fires can be readily attached to boilers which are found to be small for their requirements, and is especially useful under boilers which need forcing for short periods only. Fig. 145 shows a form of jet-blower for a brick-set boiler, and Fig. 146 for a furnace-flue.

Fans. Fans are used to mechanically control the currents of air, so that the intensity of the draft may be varied at the will of the operator and be independent of all foreign conditions.

The fans most used are of the centrifugal or peripheral discharge type. The velocity of air discharged is practically the

same as that of the tips of the blades and the pressure of the air will correspond to that velocity. The work performed is ex-

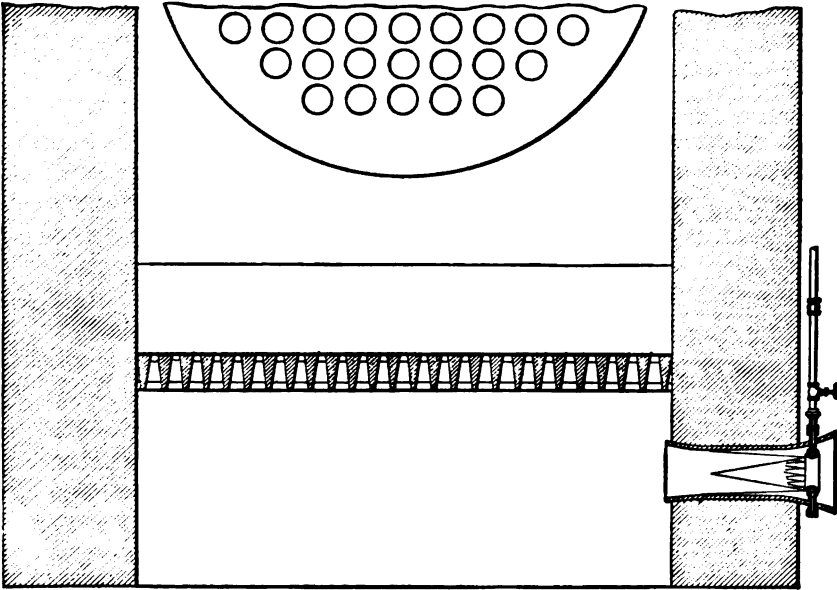


FIG. 145a.—Section of Fig. 145.

pressed by the product of that pressure times the distance through which it acts. Thus:

Let W denote the work performed in foot-pounds per second,

- | | | | |
|-----|---|---|--|
| p | " | " | pressure per square foot in pounds, |
| a | " | " | area of discharge in square feet, |
| v | " | " | velocity in feet per second, |
| d | " | " | density of a cubic foot of air in pounds = 0.0764
pounds at 60° F., |
| h | " | " | head in feet, |

then

$$v = \sqrt{2gh} = \sqrt{2g \frac{p}{d}}, \quad \text{and} \quad p = \frac{dv^2}{2g};$$

$$W = pav = \frac{adv^3}{2g}.$$

Since the fan has a working efficiency of about fifty per cent, the driving mechanism should develop a power of twice the value of W . For the proper size of fan to use under fixed conditions, the manufacturers should be consulted.*

A certain quantity of air is required to support combustion. From the formula it will be noted that the volume varies as the

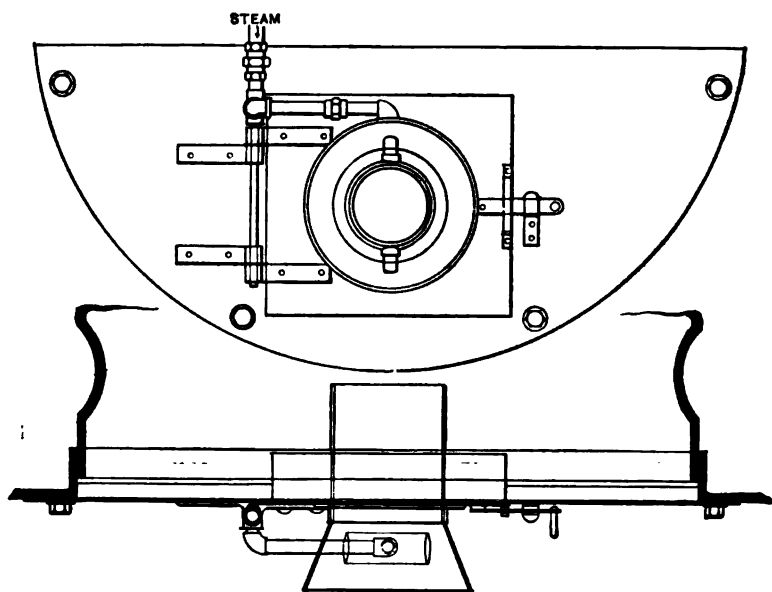


FIG. 146.—Beggs' Argand Blower, arranged for a Furnace-flue.

velocity, the pressure as its square, and the work as the cube. The best efficiency therefore is obtainable with a fan giving the volume required at as low a pressure as may be suitable. Also, the fan should be designed to comply with the exact conditions under which it is to operate. If the air is heated before it enters the fan, then the fan should be designed for use with hot air in order to save all the driving power possible.

The fan can be arranged to operate on a plenum system by having a closed ash-pit or a closed fire-room, or on a vacuum system by sucking the products of combustion through the boiler

* In Appendix B, a method is given for calculating the capacity and horse-power of peripheral discharge fans.

and discharging them into the stack, generally called "induced draft."

Closed Ash-pit System. The fan in this case forces the air into the ash-pit, all openings from which are closed, so that the air can only escape through the grate.

There is a strong tendency to burn holes in the bed of fuel by local combustion. The air then escapes through these holes without properly performing its object. As a check to this tendency, the entering air should be distributed as much as possible, according to the design of the ash-pit.

This is a very simple arrangement and one always easy to install. Economy is increased if the air is heated previous to its entrance into the ash-pit.

If the blast is strong and not shut off before opening the fire-door, the flames are apt to blow out. Automatic devices to control the draft can be attached to the doors, some of which work well, but all are liable to derangement, especially those under the severe service of marine boilers. The blast should be such as to give the required volume of air without relying on excessive pressure or velocity, a fault too common in many plants.

This system, however, is dirty, as the dust and ashes are blown outward unless a tight front is provided. When properly installed and proportioned (with furnace-flue boilers this is often difficult to accomplish) the closed ash-pit is probably the most generally satisfactory system of mechanical draft.

Closed Fire-room System. With this system the boiler is set in an air-tight fire-room, and the air blown in so as to maintain the required pressure. The air escapes into the ash-pit and through the grate. Entrances into the fire-room must be provided with double doors or air-locks, which complicate the plant for stationary purposes. The system, therefore, is hardly practicable for large boiler-rooms, and is chiefly limited to marine use.

This system also has a tendency to burn local holes in the fuel, but in a somewhat lesser degree than is found with the closed ash-pit, due to the better and more even distribution of the air. The system is, however, free from the annoyance of dirt, since all leakage is inward. When the fire-door is opened the cold air rushes in and has an injurious effect on the hot boiler plates. This can be remedied by automatic devices attached to the fire-door, so

that when open they close a damper in the uptake or breeching. These attachments do not always work satisfactorily. The same effect is attained by training the fireman to close the damper by hand before opening the door, although a trained man cannot always be implicitly trusted.

Both the closed ash-pit and closed fire-room systems tend to cause leaks between the tubes and tube-sheets of fire-tubular boilers. This difficulty is partly obviated by the use of protect-

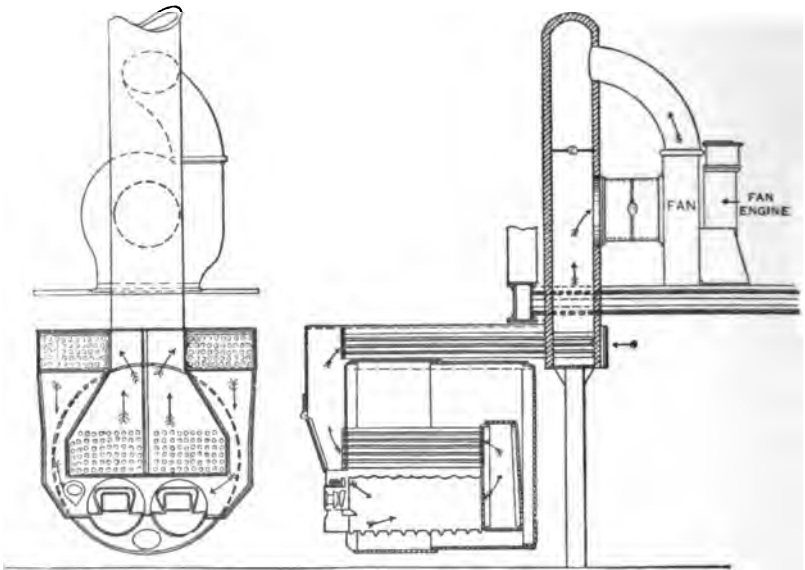


FIG. 147.—Induced Draft—Ellis and Eaves' System.

ing ferrules. The tube-sheet is often made too stiff, so that leaks are caused by want of flexibility to let it accommodate itself to the expansion and contraction of the tubes. This expansion and contraction is considerable and sudden with mechanical drafts due to the inrush of cold air through the open fire-door. As a rule the tube-sheet cannot be too flexible, the only limitation being safety.

Induced Draft. In this case the fan is so located as to suck the products of combustion through the boiler, by having the flue, uptake or breeching lead to the inlet of the fan. The fan then discharges back into the stack. This arrangement is generally

designed with a by-pass direct to the stack, which can be used in case of accident or when mechanical draft is not required. Fig. 147 illustrates a system of induced draft.

The operation of this system closely resembles the effects produced by chimney draft, only it is much more intense and capable of greater range. The objection is the deterioration of the fan and fan journals, caused by having to handle the hot gases, but many improvements have lately been made. This system has the least tendency to blow holes in the fire, and it will therefore maintain a more uniform combustion over the grate.

By simple arrangements this system readily adapts itself to the heating of both the air and feed-water, since they can be warmed by the heat in the gases without affecting the draft beyond interposing a slight frictional resistance, which can be reduced to a minimum by careful designing. Such heaters also reduce the temperature of the gases before they have to enter the fan.

CHAPTER XII

INCRUSTATION

Scurf. Fur. Sludge. Scale. Conductivity. Solid Matter in Water. Analysis of Scales. Behavior of Lime and Magnesia Salts. Scale Prevention. Blowing-off. Chemical Agents. Mechanical Agents. Galvanic Agents. Surface-condensing. Heating and Filtering. Internal Collecting Apparatus. Manual Labor.

EVEN with pure waters, a certain amount of incrustation will collect on the inside of steam-boilers. This incrustation is called "scale," "scurf," "fur" or "sludge," according to the character of its formation.

When this scale is thin, not exceeding $\frac{1}{8}$ inch, it frequently acts as a preventer of corrosion with waters that readily attack the metal. When it collects in thicker quantities, the plates become overheated and the water spaces choked. There are few satisfactory tests on the obstruction to the passage of heat by scale. When hard and dense, it offers greater resistance than when porous. More depends on the quality than on thickness as a barrier to the transfer of heat. This loss of heat is especially great when the deposits are of an oily character. The Devonport experiments, March, 1893, carried out by the British naval authorities, showed a loss of 11 per cent in efficiency due to a thin coating of grease alone.*

The tendency to cause local overheating is greatest with irregular deposits of varying thickness. This overheating is aggravated by the presence of oil and grease, which assist in making these irregular deposits. In fact a piece of greasy waste left in a boiler will cause a bulging crown sheet as quickly as a comparatively thick formation of scale uniformly distributed.

The selection of type of boiler to be adopted is often dependent upon the quality of feed-water available. With hard or bad-

* Transactions Am. Soc. Naval Engineers, Vol. IV, 1895, page 782.

scaling waters, the boiler should be of a type that can be quickly cleaned and examined. When bad water only is available, then the simplest type of boiler is to be preferred, even though less efficient in evaporating power.

Nearly all waters contain solid matters in solution, which become precipitated by elevation of temperature and are left in the boiler when the water is evaporated. This deposit, unless removed from time to time, will collect on the hot surfaces and, becoming baked, will form incrustation.

The quantity of matter thus held in solution is generally between 20 and 40 grains per gallon, but often exceeds 200 grains. To appreciate the effect, imagine a boiler evaporating 3000 pounds of water per hour. This means an evaporation at the end of four weeks, of six days of ten hours each, of 720,000 pounds, or 86,746 gallons. Assuming the water to contain 20 grains per gallon, this amount would carry into the boiler and leave as scale 1,734,920 grains, or 247.8 pounds. Taking the specific gravity of the scale as 3, this quantity would be equivalent to 1.32 cubic feet, or enough to cover 750 square feet of heating surface with scale 0.02112-inch thick, or over $\frac{1}{4}$ -inch per year. In addition to the substances in solution, nearly all waters carry clay and other earthy matters in suspension, which will greatly increase the above figures.

The quantity of solid matter contained in any water is less of an indication of its fitness for boiler use than the quality of such matter. Thus a given quantity of such salts as carbonate or chloride of sodium would be of small moment compared to an equal quantity of salts of lime. It is unfortunate, however, that the most available waters usually contain the latter salts, while the former are the exception.

Most waters contain sulphate of lime, bicarbonate of lime and carbonate of magnesia, together with other impurities of lesser importance, such as iron, soda, silica, alumina and organic matter. These impurities are deposited in the following order: First, carbonate of lime; second, sulphate of lime; third, salts of iron and magnesia; forth, silica, alumina and organic matter; fifth, chloride of sodium (common salt).

When formed, boiler scales will differ very widely in their chemical analyses. As an example, however, the following analyses

by Professor Lewes may be taken as illustrative of three kinds of water.

BOILER SCALE

	River.	Brackish.	Sea.
Calcium carbonate.....	75.85	43.65	0.97
Calcium sulphate.....	3.68	34.78	85.53
Magnesium hydrate.....	2.56	4.34	3.39
Sodium chloride.....	0.45	0.56	2.79
Silica.....	7.66	7.52	1.10
Oxides of iron and alumina.....	2.96	3.44	0.32
Organic matter.....	3.64	1.55	Trace
Moisture.....	3.20	4.16	5.90
	100.00	100.00	100.00

Bicarbonate of lime is held in solution by an excess of carbonic acid. As the water becomes heated this carbonic acid is driven off, and carbonate of lime falls as a precipitate, so that under a temperature of about 212 degrees F. it is scarcely soluble, and is said to be insoluble at 290 degrees F. Sulphate of lime also becomes less soluble as the temperature rises above 100 degrees F., and is said to be insoluble at 290 degrees F.

These two salts, therefore, are always precipitated in the boiler when the pressure reaches about 43 pounds above the atmosphere.

Carbonate of magnesium behaves in a similar manner, but it is generally found in very much smaller quantities.

The carbonates of lime and magnesium deposit in a fine white powder, making a more or less whitish sludge. These particles are so light as to be often carried by the steam into the pipes and even into the engine. These carbonates will remain as a soft sludge for some time, but will finally become hard under the baking action of the heat, as is often the case when boilers are blown off while hot. The sulphate of lime, however, precipitates so as to form an amorphous crust, which becomes hard under the action of the heat.

On becoming insoluble, all the precipitates remain for a time in suspension, and are carried around with the convection and ebullition currents until finally they settle on the plates and tubes in the quieter parts of the boiler. After ebullition ceases, they of course settle on all parts of the heating surface. The mouth of the feed-pipe often becomes badly furred, since it forms a quiet

place for the precipitate to lodge. Also the end of the blow-off may become choked for the same reason. The feed-pipes may become choked by the deposit forming as the fresh feed becomes suddenly heated, thus closing the pipe and preventing further entrance.

When oil or grease is present in the boiler, a sticky, heavy paste is formed which falls and fastens to the nearest surfaces, where it is quickly baked hard into a firm scale.

The fracture of a piece of boiler scale usually exhibits a series of layers varying in thickness and color, and the fracture may be partly crystalline, but is generally amorphous in character. Between the layers is often found soft strata of earthy matters, probably deposited while the boiler was not steaming.

The surface of the incrustation next to the metal is often black, while the surface of the metal is soft and corroded. This is due to the action of the iron salts and chloride of magnesium when present.

Chalybeate waters usually are highly injurious, and these salts of iron are detected by the reddish color of the water as it leaks out through the cracks in the scale.

Scale Prevention. Pure water, of course, should be used, but as this is not always possible, the following treatments will delay the formation of the scale and facilitate its removal.

First—Blowing off a portion of the water at regular intervals, with more or less frequency as the water is more or less bad.

This is always an easy method and consequently one of the most common practised. Many land boilers have only one blow-off, and that at the bottom. Its effect is, therefore, not general. In order to broaden the effect of the bottom blow, the pipe is sometimes extended along the bottom of the boiler and this extension part is perforated. With this arrangement the blow-off should be operated often enough to keep the pipe free from furring. If the water is very bad in scale-forming qualities, this internal pipe cannot be recommended.

Boilers should have also a surface or scum blow, which will remove some of the lighter particles, such as carbonates of lime and magnesium, oil and grease. The internal end of the surface blow is frequently fitted with a bell or trumpet-shaped mouth-piece. When the internal arrangements of a boiler are complicated,

greater difficulty is experienced in washing and cleaning it out, and such complications often are more objectionable than the good they may do in freeing from scaling matter.

Blowing off will not prevent scale, but its use may delay a thick formation and the frequent shutting down of the boiler for a more perfect cleaning. The quantity of hot water wasted by blowing out creates an expense, which in some instances would repay the introduction of some more improved method of purification.

Blowing off is most effective when the impurities are uniformly distributed, as the chloride of sodium in sea water. On the other hand, heavy precipitates are less affected by blowing off a portion of the water, and in consequence it is better with them to blow off more frequently and less in quantity at each time. Thus, for extracting magnesium and lime salts, the blows are opened in many instances once an hour, while only ten to twenty gallons are blown out at a time.

Second—Introduction of Chemical Agents to dissolve the scale. A great variety of chemical agents have been used with varying results.

The one most commonly used, being both the cheapest and the most effective for general results, is the common soda of commerce, bicarbonate of soda. It acts well in preventing and removing scale resulting from both the carbonate and sulphate of lime. The reactions are as follows: The soda and lime exchange their acids, forming sulphate of soda and carbonate of lime. The sulphate of soda is very soluble, while the carbonate of lime, freed from all excess of carbonic acid, precipitates as a light, flocculent precipitate and will not form a hard crust unless allowed to bake. The bicarbonate of lime in the feed-water is in like manner precipitated, since any free soda unites readily with the freed carbonic acid. This carbonic acid, taken by the carbonate of soda, is again liberated by the heat and the soda is free to act once more.

The soluble sulphate of soda and the deposited carbonate of lime, which is in the form of sludge, can be blown out from time to time, according to the quantity formed; or in case of large quantity, the carbonate of lime can be washed out after the boiler has been allowed to cool slowly and gradually. Since the precipitate of carbonate of lime is light and flocculent, large quantities float as

a scum when the boiler is quiet, and much of it may then be removed through the surface blow.

If the precaution of cooling off the boiler slowly be not attended to, or if the boiler be blown off while the shell and setting are still hot, the sludge may become baked hard, since it is always accompanied with more or less of the sulphate of lime.

It is found in practice that a small quantity of soda will act with good results on large quantities of feed. If soda be introduced in too large a quantity, it is apt to promote priming with all the dangers as well as the inconveniences that accompany it.

The best method is to connect the feed-pump or injector to the soda-tank, so that at regular intervals a supply of soda can be drawn. The proper amount is determined in each case by experience, but ordinarily varies between one and two pounds per day for the average boiler. The least quantity that is effective is all that is required.

The soda does not injure the boiler unless it is impure and contains acids. Soda will neutralize the acids in the feed-water and will greatly limit its natural corrosive action. Soda will also dissolve any grease that may be in the boiler, and it is often used in new boilers to cut the oil left from the process of manufacture. Both grease and soda encourage priming, so when grease is present a frequent use of the surface blow is recommended.

Besides soda bicarbonate, other chemical agents are used, but few with such generally good results and many with more or less injurious action on the boiler and its fittings. Such other agents are soda ash (an impure form of the carbonate), caustic soda, potash, chloride of barium, tannic acid, sal ammoniac and compounds of arsenic.

Third—Introduction of Mechanically Acting Agents. A great variety of substances can be introduced into boilers with the object of coating the particles of deposit, thereby decreasing their cohesion and adhesion, thus preventing them from forming into a hard mass.

The substances act either by coating the particles with a glutinous covering or by settling among the particles by interposition.

Among the first class are kerosene oil, petroleum, sugar-cane juice, molasses, moss, seaweeds, potatoes, tallow and starch.

Kerosene is considered better than petroleum, and both act well when sulphates are present. Most of the other substances contain acetic acid and act best when the sulphates are absent. The acetic acid will attack the boiler-plates and the organic compounds will form scale with any sulphates that may be present.

Among the second class are clay and some kinds of wood, such as mahogany, logwood and hemlock bark, in the form of powder or fine chips, which may be introduced with the feed. The substance is expected to settle along with the deposit and facilitate its removal by preventing a solid-mass formation. These substances, however, increase the solid matter in the boiler, and their distributed settlement throughout the mass of scaling deposit cannot be relied upon. They are, therefore, as a class better avoided.

Some of the anti-incrustation compounds act according to one or both of the above methods. In general they should be used with caution, as many of them will cost the steam-user more in the reduced life of the boiler than they save in coal.

Fourth—Galvanic Agents. Sheets of zinc have been used in the inside of boilers, and the results claimed have been more or less favorable. Their use is principally in connection with salt water and their efficacy appears to be due to the action of the alkaline chloride of zinc on the salts forming the scale.

Fifth—Surface-condensing. The exhaust steam is condensed in a surface condenser, and the hot water saved and pumped back into the boiler, fresh water only being used to supply losses from leaks, safety-valve and whistle. This method is common in marine practice, but not so frequently used on land on account of the scarcity or expense of water for condensing or cooling purposes. Since all the oil and grease used in the engine cylinders pass into the condenser, it is necessary to filter the water before it is returned to the boiler. This is done by some filtering material, as sponges, straw, salt-meadow hay, excelsior, flannel. etc.. arranged in a variety of ways.

If insufficient cooling water be used, the elevation of temperature will cause a deposit to form on the condenser tubes, especially when much bicarbonate of lime is present. No deposit takes place when salt water is used for cooling purposes, because the salt cannot be precipitated by elevation of temperature. If, however,

the condenser leaks, the salt water would enter the feed and deposit in the boiler.

Sixth—Purification by Heating and Filtering previous to entrance into the boiler. It is now becoming quite common to heat the feed-water when bad under pressure in a closed vessel, warmed by exhaust or live steam. Under this treatment the lime, magnesia, etc., are nearly all deposited in this external collecting vessel, which should be so arranged as to be easily and quickly cleaned.

Feed-heating coils are sometimes arranged in the smoke-flue, but if the water is bad this method should not be encouraged, as such pipes are not easy to get at unless the boiler be shut down or a by-pass smoke-flue provided.

It is evident that this treatment does not prevent incrustation, but merely allows the deposit to form in a special vessel separate from the boiler, in which the scale is not liable to be baked hard.

Many of the anti-incrustation compounds might be used in one of these vessels more advantageously than in the boiler itself, although they cannot be recommended even in such a connection. These external vessels form a feed-heating apparatus, since the water must be heated to a high degree before the vessel becomes effective as a purifier.

When the feed contains large quantities of matter in suspension, it is well to filter it through sand, pebbles, etc., before pumping it into the boiler. Such filters should be arranged to be easily cleaned, which is generally done by passing the wash water through it in a reverse direction and allowing it to waste.

Seventh—Internal Collecting Apparatus. Sometimes curved plates and troughs of various shapes are arranged inside of the boiler with some success. It is intended that the major part of the deposit will form on these surfaces, which should be made to be easily removed, cleaned and replaced.

Thus the feed can be arranged to discharge into a trough placed just above the water-line, and in which the water will be comparatively quiet and free from ebullition. The surplus feed will overflow the edges. Most of the lime and magnesia salts will deposit in the trough on account of the increase of temperature. Such an arrangement works best with the sulphates and heavy deposits.

Internal-collecting apparatus when complicated interferes

with the proper inspection and cleaning of the boiler itself, and it is better to have the collecting device outside of the boiler, in a feed purifier of good design, as described under the sixth treatment.

Eighth—Removal by Manual Labor. In erecting a new plant where bad water only is available, the greatest care should be taken to select a boiler that has a strong circulation over crown sheets and parts subjected to great heat and difficult to clean. Patented devices to attach to boilers to strengthen their circulation are not always reliable.

After the incrustation has formed to a certain thickness, depending on local circumstances, the boiler is blown off and the scale detached and removed by manual labor. This chipping of the scale is often a difficult work; and when it is hard, the men, if careless, are apt to injure the surface of the plates and the ends of the rivets. Dents or abrasions thus formed in the surface of the metal afford good opportunity for the corrosive action of the feed-water.

Special tools often are made to conveniently reach difficult places; and chains are sometimes inserted in the water-legs, so as to wear off the scale by attrition, as they are drawn back and forth.

When a boiler is suddenly blown off while hot and then refilled with cold water, the scale is frequently cracked and loosened, and perhaps thrown down by the contraction of the plates. This is, however, a very injurious process, and should never be permitted. It simply saves the cost of a little manual labor.

On the other hand, if the boiler be allowed to cool gradually, and then be blown off after standing full of water for three or four days, the deposits are not apt to be baked hard, and some of the salts may be softened or redissolved. The objection to this latter treatment is the long time the boiler is out of service, but as all important plants should have a spare boiler, this objection has less practical value than would at first appear.

If, however, time is an important feature, the cooling of the boiler may be hastened by pumping in fresh water as the hot is slowly blown out. This treatment, when done slowly, will uniformly cool off the boiler, and will also keep the bottom blow from becoming clogged or choked with sludge, as may happen if the boiler be allowed to stand for some time.

CHAPTER XIII

CORROSION. GENERAL WEAR AND TEAR. EXPLOSIONS

Corrosion. Wasting. Pitting and Honey-combing. Grooving. Influence of Air and Acidity. Galvanic Action. Zinc Plates. External Corrosion. Dampness. Wear and Tear. Idle Boilers. Explosions. Stored Energy.

FROM the moment the boiler is finished, it gradually becomes weaker, due to the destroying forces that are continually acting upon it. These forces are of both a chemical and a mechanical nature, and keep up their influences with an ever-increasing rapidity.

Corrosion, which takes place both internally and externally, is the most serious and subtle cause of weakening to which a boiler is subjected.

Internal corrosion is generally in the form of uniform wasting, of pitting and of grooving.

This Wasting Away of the plates is seldom so uniform in its effects as rusting, but it usually covers large patches of surface. It is produced by some chemical action of the water on the plates. It is more or less easy of detection. Sometimes it produces "bleeding," that is, the scale and plates in the vicinity are streaked with red. When wasting is very extended, it is apt to pass unnoticed on a hasty examination, but it is always revealed by drilling the plate and calipering. This precaution of drilling should always be resorted to with old boilers. There are no rules to guide the inspector in searching for wasting, and it does not appear to take place in two boilers alike. However, the water-line appears to be especially liable to attack in boilers that stand quiet for a long time. For this reason boilers should not be allowed to stand idle when only partly filled with water. Sometimes this corrosion is

hidden under a thin shell or crust of metal, when its presence can be detected by a testing hammer.

Pitting, or honey-combing, is in general well defined. It consists of small depressions of varying shapes and forms. When the depressions are extended the effect is more like wasting. It is not limited to any parts of the boiler, and it often appears in the steam space. The depressions are sometimes filled with a fine powder, being a mixture of iron, silica and other earthy matters, and a small percentage of oil. The depressions are occasionally covered with a hard crust like a blister, but more frequently are open with sharp edges.

M. Olroy, a French engineer, thus states * the results of his investigations of the pitting of boilers: "Pitting is particularly likely to occur if a water very free from lime is used in clean boilers. When a boiler forms one of a battery and is kept standing for a long interval, the top of the boiler is liable to pitting. Steam finds its way into the boiler, and condensing upon the top surface, causes bad pitting there. Pure water containing no air does not harm, and steam alone will cause no pitting unless it contains a supply of air. The Loch Katrine water of Glasgow, which causes pitting in clean boilers, contains much gas. The water from many of the lakes in America also produces the same effect. With distilled water the boilers usually remain quite bright. Feed-water heaters often suffer badly from pitting, particularly near the cold-water inlet, and in boilers the parts most likely to be attacked are those where the circulation is bad, especially if such portions are also near the feed inlet.

"In locomotives the bottom of the barrel and the largest ring is most frequently attacked. The steam spaces are generally free from pitting, unless the boiler is frequently kept standing with water in it. As the water evaporates or leaks away, pitting is liable to occur along the region of the water-line, a part which in a working boiler is generally free from attack, unless the longitudinal seam is near that point and forms a ledge where the moisture can rest.

"Pittings take the form of cones or spherical depressions, which are filled with a yellowish-brown deposit consisting mainly of iron oxide. The volume of powder is greater than that of the metal

* Engineering, 19 October, 1894, Vol. LVII.

oxidized, so that a blister is formed above the pit which has a skin as thin as an egg-shell. This skin usually contains both iron oxide and lime salts and differs greatly in toughness. In many cases it is so friable that it breaks at the least shock, falling to powder, while in other cases the blister detaches itself from the plate as a whole.

"An analysis of the powder in the pits shows it to consist of peroxide of iron, 86.26 per cent; grease and other organic matter, 6.29 per cent; lime salts, 4.25 per cent; water, silica, aluminum, etc., 3.20 per cent. The skin over the pits was found to contain calcium carbonate, 38 per cent; calcium sulphate, 12.8 per cent; and iron oxide, FeO , 32.2 per cent, and about 8.5 per cent each of magnesium carbonate and insoluble matter."

Grooving occurs along the edges of laps, angle-bars, stays and doubling-plates. It is caused primarily by too great a stiffness, resisting the expansion and contraction. This stiffness compels the play or breathing of the boiler to take place locally, similarly to bending back and forth a thin plate with the hands. It is sometimes initiated through injury to the plates by a careless use of the calking chisel or cleaning tool. In a crack once started, grooving increases rapidly, due to the corrosive action entering deeply into the plate and exposing fresh surfaces.

When cylindrical boiler-shells are too firmly seated on their foundations, the expansion may promote grooving by causing the plate to buckle near the lap and butt-straps, especially those of the transverse or ring seams. If the shell be not perfectly cylindrical, it tends to become so under pressure, and the longitudinal seams are similarly affected.

Grooving may also be caused by lack of stiffness. For instance, in vertical fire-box boilers, grooving is liable to occur at the mud-ring when this ring is not made of sufficient depth to resist the upsetting action caused by the expansion of the fire-box sheet being greater than that of the outer shell sheet.

To prevent grooving of the head or end plates at the flange or angle where they join the shell, tubes, flues and stays should not be located too near the shell; the flange should be turned with an easy radius, about two and a half thicknesses, and the head should be made as thin as safety will permit.

Internal corrosion appears to be caused by acidity and air in

the feed water, and perhaps in special cases to some form of galvanic action. Such galvanic action has been counteracted by the use of zinc plates. The electric couple separates the hydrogen, which passes to the steel and then escapes into the steam space, while the oxygen goes to the zinc. The proportions found requisite to insure protection are about 1 square foot of zinc to 50 square feet of heating surface in new boilers, and the same quantity of zinc to 75 or 100 square feet of surface in older boilers. The zinc plates should be about 10 inches by 6 inches by $\frac{1}{2}$ -inch thick. The contact between the zinc and steel must be a good metallic contact in order to be effective. Usually the zinc is bolted to convenient stays or to studs formed for the purpose on the combustion-chamber. The contact surfaces should be bright and clean, and the nuts should be well screwed down to prevent scale forming between them. Zinc slabs will last from two to three months. The zinc is often carried in metal baskets to catch pieces that break off from time to time.

The varied appearance and location of corrosion may be caused by the lack of either physical or chemical homogeneity in the metal. In order to prevent internal corrosion, the causes should be studied and neutralized as far as possible. The materials of the boiler should be as homogeneous as possible; the feed-water should be kept slightly alkaline by the use of soda and be free from air. A thin coating of scale will often act as a protective covering. When corrosion has attacked a surface that is not directly exposed to the fire or hot gases, it will often be found beneficial to clean the spot and wash it well with soda, and paint it with a thin layer of Portland cement. Wherever boilers are liable to excessive corrosion, they should be most carefully examined at regular intervals. This is true for external as well as internal corrosion.

External Corrosion is just as active as internal corrosion, and in many ways is more dangerous, since it is not so often suspected and since boilers are frequently so set as to be difficult to properly inspect.

The forms are similar to those of internal corrosion. It is generally produced by leaks or dampness, and the drippings from fittings and gauges. Ashes will rapidly corrode any metal against which they lie.

A brick setting should be kept away from contact with the

shell, and care should be exercised not to cover a longitudinal seam. In fact all seams should be exposed, as far as the design will permit, that leaks may be detected at once. The brick setting should be carried to within about $\frac{1}{4}$ -inch of the shell, and this space be packed with fire-clay or asbestos fibre. Many boilers are so set that their tops are covered with a brick arch. It would be better, when possible, to construct the side and back walls high enough to retain a heavy layer of clean dry sand. This sand can always be pushed aside for the purpose of inspection. When felting, asbestos, magnesia or similar coverings are used, they should be laid so as to touch the boiler without an air space, that any leak may soak through and become visible. All fittings or attachments covered by brickwork, such as blow-off connections, should be avoided, as they cannot be inspected except by the removal of the masonry.

General Wear and Tear weakens a boiler by a more or less gradual process, with activity increasing with age and lack of care. Repeated expansion and contraction, especially when sudden and local, are principal factors of encouragement. When boilers are fired up too suddenly, certain parts are heated faster than others, and undue stresses are brought to bear which often cause buckling and straining. This effect is noticed on the transverse or ring seams, which show a tendency to groove under the laps or butt-straps. Length of shell, therefore, may be an element of weakness, although the metallic strength to resist bursting has been shown to be independent of length.

Boilers often leak after having been tested and made tight. This may be due to a variety of causes, but usually can be traced to severe handling or to excessive variations in temperature. Sometimes the cold feed enters so as to impinge against a hot plate, tube or riveted seam, and leakage is sure to result. Leaky rivets should not be calked too much. It is better to cut out the rivet in fault, ream the hole fair and insert a new rivet to fit the new hole.

Idle Boilers should receive attention. When boilers are laid off, care must be taken to arrest the actions described in this chapter.

The outside should be cleaned and painted with a good metallic paint, applied directly to the cleaned and dried surface. If the

boiler be covered by lagging, the lagging should not be allowed to absorb moisture from the atmosphere.

On the fire side, the soot and ashes should be thoroughly removed and the surface cleaned. These surfaces should then be kept dry and not exposed to damp air. Fresh lime in pans or trays, renewed as required, will absorb the moisture in the air. Occasional small fires of tarred wood will be beneficial, as the heat will dry the metallic surfaces and the resinous condensations from the thick smoke will cover the tubes and shell with a protective coating.

On the water side, corrosion may be active at the water-line if the boiler be left partly full. Idle boilers should, therefore, be entirely dry or completely filled with water. If the laying off is for a short time only, it is a good plan to fill the boiler with water made alkaline by a little soda. If for a long period, it seems best to empty the boiler and dry out the inside by a small fire built in a pan, which can be inserted through the lowest manhole. The manhole and handhole covers can be put back and the boiler made tight so that the oxygen will be consumed by the fire, or the covers can be left off and lime in trays used to absorb any moisture.

Explosions occur when the steam pressure exceeds the resisting strength of the metal structure.

In a well-designed boiler the parts are of approximate equal strength throughout. It is good practice to so design a boiler that those parts shall have an excess of strength which are expected to suffer most rapidly from corrosion or wear and tear. Then as the boiler advances in age, the various parts become more nearly equal in strength.

Should a boiler become weakened and a rent occur, the steam pressure will be suddenly reduced, thus releasing the heat stored in the water. The water instantly flashing into vapor probably accounts for the great destructive effects produced by an explosion.

While the rent primarily occurs at some weak spot, the fracture may not and seldom does follow a line of structural weakness. The new forces set up at the instant of explosion no doubt account for this phenomenon.

Imagine a boiler to contain 60 cubic feet of water and 10 cubic feet of steam under a pressure of 120 pounds per square inch above the atmosphere. Since each cubic foot of water weighs 57.3 pounds

and each cubic foot of steam 0.301 pound, the total heat contained above 212° F. in B. T. U. would be:

$$\begin{array}{r} 60 \times 57.3 \times (321.4 - 180.7) = 483,726 \\ 10 \times 0.301 \times (1106.1 - 1074.2) = \quad 96 \\ \hline 483,822 \text{ B. T. U.} \end{array}$$

and $483,822 \times 778 = 376,413,516$ foot-pounds.

Consider the problem in another way. The work could be expressed by the change of volume of the water into steam times the atmospheric pressure plus the expansion of the steam to atmospheric pressure, or

$$3.323 \times 144 \times 14.7 \times 60 \times 57.3 = 24,183,201$$

$P_1 V_1$ hyp-log r

$$\begin{aligned} &= 144 \times 135 \times 3.323 \text{ hyp-log } \frac{26.64}{3.323} \times 60 \times 57.3 = 462,284,027 \\ &\text{Work in foot-pounds} = 486,467,228 \end{aligned}$$

Neither of these assumptions correctly measures the power expended, but as exact data are always missing at time of explosion, no accurate calculation can be made. The figures illustrate two facts: first, that there is sufficient energy in the boiler to create the destructive effects attributable to an explosion; and, second, that the energy stored in the steam is very much less than in the hot water. All things being equal, the damaging effect by explosion of water-tubular boilers will be less than of fire-tubular boilers of equal rating, since the former contain a smaller proportion of water, and since extra time will be required for complete release, because the bursting part is small.

Failures of boilers are usually due to wear and tear, produced chiefly by expansion and contraction, to corrosion, to overheating and to carelessness. Overheating may be caused by low water or by scale or grease. Important fixtures, such as main stop-valves, may become attacked, or the main steam-pipe may be burst by water-hammer, thus causing a sudden release of pressure, which, if quick enough, may be followed by an explosion.

Explosions of sectional boilers of the water-tubular type do not produce effects so destructive as those created by fire-tubular boilers. Most water-tubular boilers could blow out a tube and still

not explode, as the time required to release the pressure is an all-important element. This property is one of the chief claims favorable to that class.

When an explosion does occur, it is frequently very difficult to determine the cause, and hasty judgment should always be withheld. A good piece of metal may show a poor quality of fracture on account of the suddenness of the rupture. Opinion as to the quality of the metal should only be given after a close and careful analysis of physical and chemical tests.

The best way to prevent explosion is to employ intelligent labor and not neglect proper and regular inspection.

CHAPTER XIV

CHIMNEY DESIGN

Object. Selection of Height. Compare Cost of Stack with Mechanical Draft. Individual Stacks in Lieu of One Large Stack. Self-supporting and Non-self-supporting Stacks. Wind Pressure. Batter. Brick Stacks. Section. Lining. Top. Lightning. Ladder. Leakage. Steel Stacks.

A chimney or stack is a necessary adjunct to all furnaces. It has a twofold use, namely, to create a draft or current of air through the bed of fuel, so that the process of combustion may be continuous; and to discharge the products of combustion at such elevations as to be least objectionable.

When the design relies upon natural draft, the stack must have the requisite area of flue and height to produce the flow of hot gases through it, as determined by the quantity of fuel to be burned in a given time. Owing to the ever-changing conditions, as temperatures of air and gases, grade of fuel, rate of combustion to suit variations of work, etc., the height is usually settled by experience. If there is doubt, give preference to increase of height, as too strong a draft is rather a good fault than otherwise. If on account of the cost, or for any other reason, the desired height cannot be secured, then the flue area should be made proportionately larger, so that the required quantity of gases can be discharged at a lower velocity.

With a mechanical draft, the height of stack need only be such as to obtain a suitable outlet for the gases.

Where the products must be discharged so as not to create a nuisance, no fixed information as regards height is available, but each case must be worked out from its peculiar surrounding conditions.

A good, strong, natural draft is most desirable, since it produces an even and economical combustion of coal. Its intensity

is controllable to suit varying conditions by the use of the damper. But to obtain a strong, natural draft capable of drawing the products of combustion through the furnace and boiler, as well as through a feed-water heater or economizer, means the construction of a high stack. The cost of producing this draft is evidently the sum of the repair, interest and depreciation charges against the stack. If the stack is very high and of expensive construction, natural draft is costly.

Before the final design of the stack is determined, the cost of an equivalent draft produced by mechanical means should be considered for comparison. A mechanical draft might save sufficient in first cost of stack, that the interest on this amount added to the repairs of the proposed stack may pay the charges for operation, maintenance and depreciation on the mechanical-draft apparatus.

Owing to the great cost of tall stacks, many plants are providing a number of short stacks the aggregate cost of which would be less. Furthermore, when a number of boilers are connected to one stack, some are liable to rob others of draft, a state of affairs difficult to prevent in practical operation. This can be obviated by separate stacks to each boiler or by one stack common to three boilers when such boilers are set close together. The separate stack plan does away with a long breeching or flue connection to the stack, which is often an item worthy of consideration, both as to cost and loss of draft "head." The breeching being horizontal or nearly so, permits of accumulations of soot, which may cause corrosion and presents difficulties to clean unless all the connecting boilers be shut down.

Stacks may be of two kinds, self-supporting and non-self-supporting. The latter class requires bracing against side pressures due to wind. Such braces or guys are generally of wire rope $\frac{3}{4}$ -inch in diameter, or iron rods $\frac{1}{2}$ -inch in diameter and 10 to 20 feet long, linked together like a chain.

The weight of the stack is carried by the foundation, which must have an area determined according to the bearing properties of the soil. The weight to be supported is the dead weight of the structure plus the wind load. The area or strength of the stack at any point must be capable of carrying the weight and wind stresses above such a point.

The wind acts as an overturning force and is resisted by gravity. Stability calculations should be made at frequent sections. For

a masonry stack, the resultant of the wind and weight forces should pass through the section considered not farther from the axis than the sum of one-half the outer radius plus one-quarter the inner radius. If the stack be square or octagonal, use the radii of the inscribed circles. Calculations should be made for compression and tension. The maximum compression for radial brick is taken generally at 15 tons per square foot, and the tension at $2\frac{1}{2}$ tons. The formula is:

$$\frac{\text{Weight}}{\text{Area}} \pm \frac{\text{Wind moment}}{\text{Section modulus}} = \text{Max. unit stress.}$$

In estimating the weight, only that of the stack and base should be considered. The weight of the lining should be omitted unless of a very permanent character.

To increase the stability, the sides of base can be lengthened by making the foundation on a vertical batter of one horizontal to three vertical.

It is customary to assume the wind pressure at 50 pounds per square foot of surface and as acting on the full vertical area of one side of square stacks, on three-fourths the area of vertical section of octagonal stacks, and on one-half the area of vertical section of round stacks. The point of application of the pressure is taken at the centre of area of the exposed section. It is highly probable that the centre of pressure is above that point due to the lesser velocity of wind at points low down, since the air is appreciably retarded by friction with the ground. Ample allowance should, therefore, be provided by assuming a high wind velocity.

Excepting short steel stacks that are bolted directly to the boiler, the bases should be made to spread out in order to add to the stability. Stacks should taper toward the top with a batter of not less than $\frac{1}{4}$ -inch to the foot when of brick, while steel stacks may have a smaller batter if desired. Short steel stacks are usually made parallel. If there is too small a batter, tall stacks will look top-heavy, and the required amount of batter largely depends on appearance.

Stacks are made of brick, reinforced concrete, and steel, and the foundations are either of brick, stone, or concrete. In preparing a design it will be found just as simple and cheap to make the stack graceful and pleasing to the eye as to create an ugly and stiff looking structure. All fancy ornamentation or

pattern work should be avoided as being unsuitable and detracting from the general appearance of solidity. Any ornamentation of a kind that will rapidly deteriorate should especially be omitted. A plain, simple design having easy and graceful lines is the one most appreciated. A study of the designs of chimneys as published in the engineering periodicals will well repay the trouble before completing a new design.

Brick Stacks. The round section is the most effective, but it is also the most costly. Next to it is the octagonal section, and then the square.

The round chimney is generally difficult to locate close to buildings without causing a waste of room. The base, however, can be made square, giving the appearance of a pedestal to the cylindrical shaft which is often pleasing. Round flues are theoretically preferable to square ones as offering less frictional surface, but in stacks of moderate height there appears to be little, if any, practical difference in the draft intensity. For tall stacks the round section is to be preferred, and is favored by many engineers for all cases. Under ordinary conditions, structural and esthetic considerations should settle the flue section and consequently the outside section to be adopted.

The structure should be built with good, sound bricks of uniform color, laid flush in cement mortar,* having a minimum thickness at top of $8\frac{1}{2}$ inches for common brick and 7 inches for radial brick. It should be lined with fire-brick, the lining being carried up from half to two-thirds the height in short stacks and from one-quarter to one-third in tall ones. The lining should be independent of the stack, leaving a space at the bottom tapering to nothing at the top, so that it may expand freely and be easy to remove and renew. The header bricks of the lining should project so as to touch the outer wall, but not be bonded to it. The lining may be continued up with hard-burnt brick after the fire-brick stops, if desirable. Offsets in the lining to increase its thickness should be made on the outside, and sections of the same thickness are generally from 40 feet to 50 feet high. The top section is usually about 4 inches thick, but a 4-inch section should not be over 25

*The cement may be a mixture of Portland cement, slacked lime and sand, in the proportions of 1 measure of cement, $\frac{1}{2}$ measure of lime and 3 measures of sand. The lime is added to make the mortar set slower and work smoother.

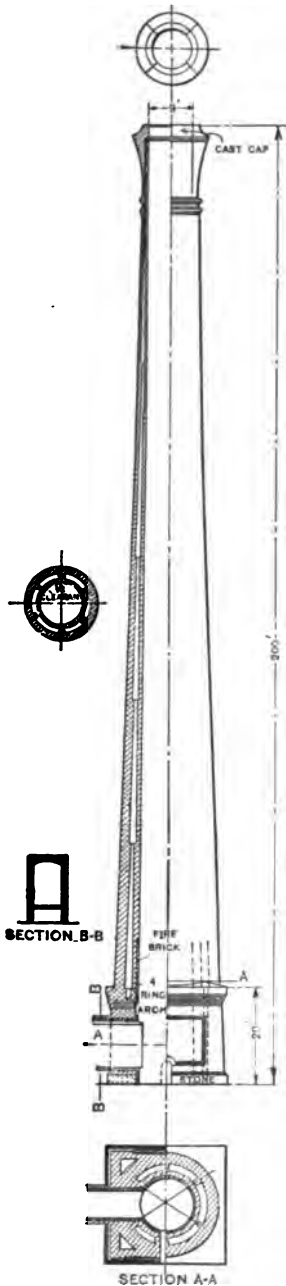


FIG. 148.—Brick Stack. Designed by E. S. Farwell.

feet high. The flue area is generally made tapering toward the top, but sometimes is of uniform area.

The top of the stack should be finished off with a capital of suitable design. It is claimed that the draft may be assisted by adopting an appropriate shape. The principle of the claim is that the upper surface should incline upward and inward, so that the air current passing over the chimney will cause a suction. The top may be capped with stone flagging or by an iron casting shaped to bond the last courses of brick. These iron caps, especially when large, are best made in sections bolted together through flanges.

Tall stacks are protected by lightning-rods. One point of $\frac{3}{4}$ -in. solid copper, sharpened and tipped with platinum cap $1\frac{1}{4}$ in. long, should be used for every 75 feet in height. The points are connected to a stout copper band. The band is connected to the ground by two conductors of $\frac{1}{2}$ -in. stranded copper cable, terminating in a coil buried at least 6 feet in a bed of charcoal. The cables should be secured to the stack by suitable brass anchors.

There should be a ladder on every chimney, made of iron rods about $\frac{3}{8}$ -inch in diameter, built into the brickwork at convenient distances apart, generally about every 16 inches. The ladder may be on the inside or outside to suit the conditions or fancy of the designer. In general, it is the more useful when on the outside.

The connecting flue or breeching may enter the stack through the side or

foundation, which is generally about one-tenth or one-eighth the height of stack. Many designs are so stiff, however, as not to

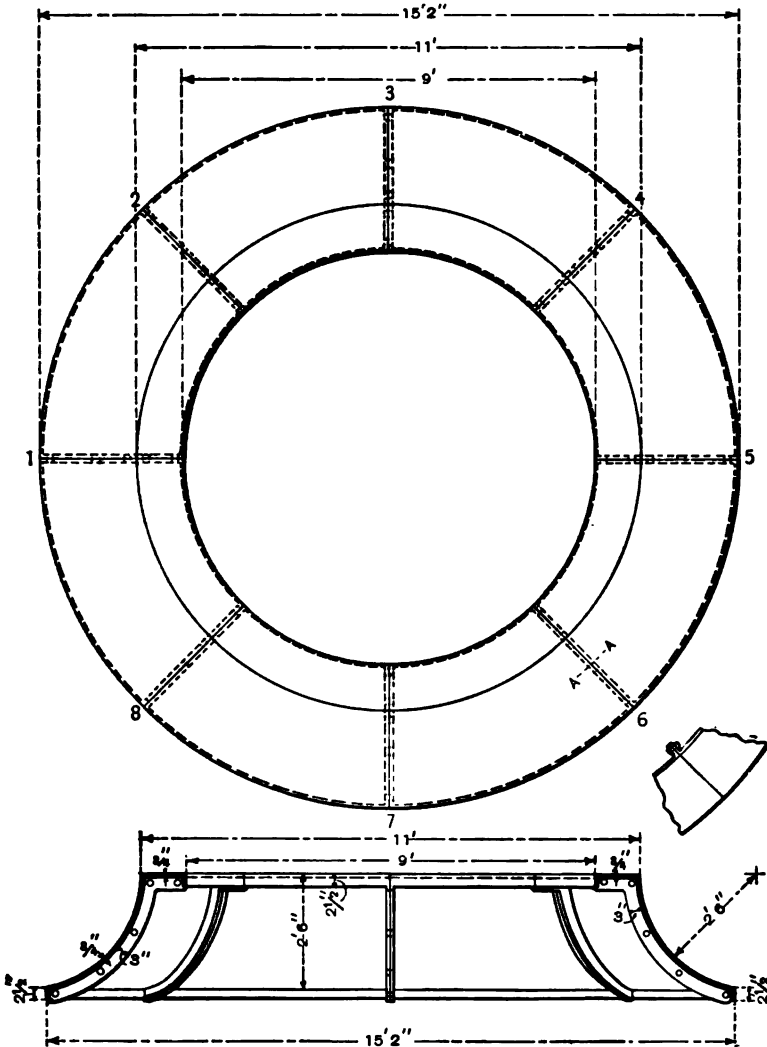


FIG. 150.—Cast-iron Cap for Fig. 148.

require support from these anchor-bolts, although it is always well to use them. The base section is generally bell-shaped, having a

bottom diameter about twice that at top of bell and a height equal to the bottom diameter.

Steel stacks should be lined with brick, as the lining prolongs the life of the metal and materially adds to the stability of the structure. In short or unimportant stacks the lining is frequently omitted.

In large stacks it is advisable to build the lining in self-supporting sections to facilitate renewals, and to prevent a general failure due to disintegration. These sections are usually made from 12 to 20 feet in height.

The remarks on the shape of the top, the entrances for flues and clean-out, and ladder are equally applicable to steel as to brick stacks.

Plates thinner than $\frac{1}{4}$ inch are seldom used,* on account of the weakening by corrosion. The size of rivet for varying thickness of plate should be about the same as for boiler-shells, but never less than $\frac{1}{2}$ -inch in diameter. The pitch of rivets may be as given in Table XVIII, although it

* Except in stacks less than 40 feet in height and not over 20 inches in diameter.

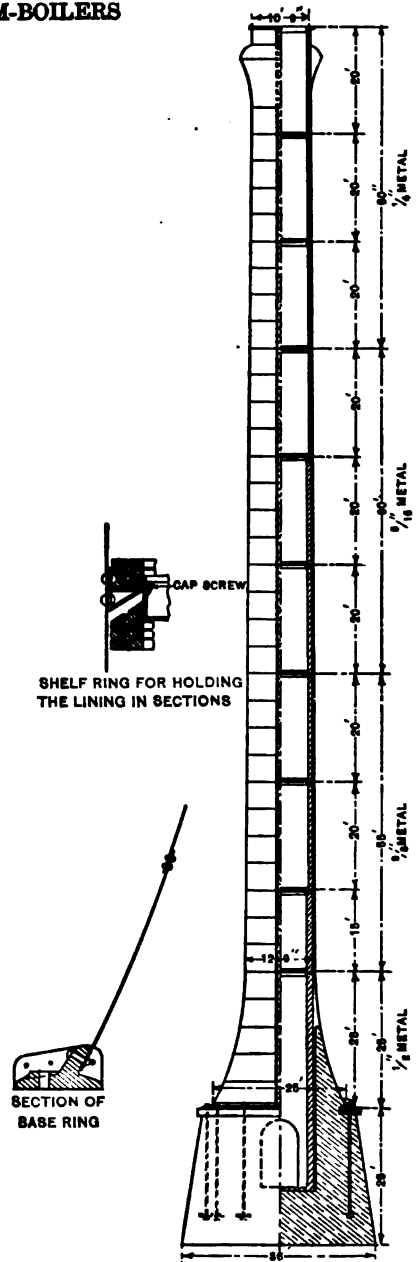


FIG. 151.—Self-supporting Steel Stack.

is usually somewhat greater. For the first 50 feet from the top the horizontal seams can be single-riveted; for the next 150 feet, double-riveted; and for any additional length, treble-riveted. The vertical seams can be single-riveted for the first 100 feet from the top and double-riveted for any additional length. The plating should be well lapped at the seams, and be calked to prevent air leaks.

In self-supporting stacks the factor of safety for the holding down bolts should be at least four, and only half the bolts should be considered as in tension at one time. These bolts should not be spaced over four feet apart, and there should never be less than four. If additional bolts are used, the total number is usually a multiple of three or four—thus four, six, eight, nine, etc.

The thickness of the plating can be proportioned by the following formulæ, the safe fibre stress being taken at 10,000 pounds per square inch:

$$\left. \begin{array}{l} \text{Stress per lineal inch} \\ \text{at any section} \end{array} \right\} = \frac{\text{Moment for wind in inch-pounds}}{\frac{1}{4}\pi \times (\text{diameter in inches})^2};$$

$$\text{Thickness in inches} = \frac{\text{Stress per lineal inch}}{10,000 \times \text{efficiency of horizontal joint}}$$

The Riter-Conley Manufacturing Co.* use a similar formula, namely, $S = \frac{M}{0.8d^2}$, but neglect the efficiency of joint and adopt 8000 pounds per unit stress if the circumferential seams are single-riveted and 10,000 pounds if double-riveted. Steel stacks as manufactured by this company have proved satisfactory, and possibly the thickness as determined by the above formulæ may be greater than required except in very exposed locations.

A brick stack is illustrated in Fig. 148, a ladder in Fig. 149, and the cast-iron cap in Fig. 150.

A steel stack is illustrated in Fig. 151.

* Of Pittsburg, Pennsylvania. Kindness of Mr. Wm. C. Coffin, Vice-President, 1903.

CHAPTER XV

SMOKE PREVENTION

Losses Due to Smoke. Public Nuisance. Smoke Ordinances. Requirements to Prevent Smoke. Prof. Ringelmann's Smoke-scales. Smokeless Fuels. Composition of Smoke. Mixing Coals. Air Admissions. Hollow Bridge. Extracts from Report by Prof. Landreth.

The study of smoke prevention is intimately interwoven with that of combustion and of boiler design. When combustion is perfect the products of combustion are practically colorless.

Smoke consists of soot or carbon in a flocculent state, mixed with the products of combustion, namely, carbon dioxide, carbon monoxide, sulphuric and sulphurous acid, water, nitrogen, ammonia, carbureted hydrogen and other vapors of lesser note. The losses due to smoke generation are not great, probably not exceeding in any case more than $1\frac{1}{2}$ or 2 per cent of the heat generated. Possibly the cost of smoke prevention with some fuels would exceed the value saved in heat. On the other hand, smoke can be taken with few exceptions as an evidence of uneconomical combustion, whereby the real losses greatly exceed the above figures.

Smoke has become such a public annoyance in closely peopled districts as to warrant the officials of many cities to pass prohibitory ordinances. Admitting that smoke is a public nuisance, the making and enforcing of preventive regulations appear to be most satisfactory when they originate through the local boards of health rather than by ordinance, although the smoke may not be in sufficient quantity as to be injurious to health. The soot or carbon is not in itself injurious, but it is a nuisance when it soils surrounding objects. The compounds of sulphur and ammonia are injurious when in quantity, but a smokeless furnace may pass off large amounts of these products. Such regulations should be carefully drawn and made to keep pace with the advances being continually

made in steam engineering. They should be on some fair-minded penalty basis, founded on what can be done commercially without inflicting too heavy a loss or expense, rather than what could be done by compulsion.

Almost any fuel can be consumed so as to produce little or no smoke. This is not the case with fuels burned in furnaces designed and set for one grade and consuming another; nor in combustion-chambers so built as to be ill adapted to encourage perfect combustion. Due to commercial changes, some steam-generating localities are using soft coals, in which hard coals were used almost exclusively some few years ago. The same boilers, same settings and same methods of firing no doubt are still largely employed, irrespectively of the changed conditions. It is not the fault of the fuel that smoke under such circumstances is produced.

The heating surfaces of the boiler rob the products of combustion of their heat, and thus reduce their temperature below that necessary for chemical union with the oxygen. If the constituent particles of the fuel have not been mixed with the incoming oxygen before this reduction of temperature, the carbon, in a finely divided state, will pass off with the draft and create smoke. To prevent smoke, therefore, the requirements are those conditions which will furnish perfect combustion, namely, a good draft, a proper mixing of the air and fuel, and a maintenance of the high temperatures until the chemical unions are completed. These conditions would be easy to obtain in a suitable furnace if it were not for the shortness of time available.

Smokeless fuels, such as oil, cannot be considered at this time as a substitute for the smoke-producing fuels, since they are not found in sufficient quantity to supply the demand and the artificial fuels are still too costly. Anthracite, unfortunately, is too expensive to compete with the soft or smoking coals. It has been shown on trial that a combination of oil and bituminous coal can be used so as to produce a practically smokeless fire, but this combination, however, has not proved very successful commercially.

Bituminous coal in selected sizes—about 3-inch cubes—can be burned practically smokeless. Smoke from bituminous coal can be reduced by mixing 50 per cent of anthracite pea or coke with the bituminous coal. The object of the mixing is to separate the bituminous coal, so that air can reach every particle.

Mechanical stokers and down-draft furnaces under suitable conditions reduce the amount of smoke visible to the eye, but do not lessen the other ingredients beyond their tendency to assist combustion. As the best-known mechanical means of firing are incapable of lessening the compounds of sulphur and ammonia, it has been suggested that the true solution of smoke prevention would be to reduce the coal to gas and then purify it before use. This solution may come in time, especially for thickly settled manufacturing districts, but cannot be forced.

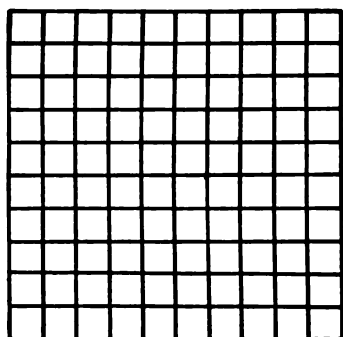
If an ordinary grate is used with bituminous coal, there should be not less than 36 inches between grate and boiler surface. A greater distance would be still better, especially when very smoky coals are used. Some grates produce smoke from a lack of air openings. These openings for smoky coals should aggregate between 50 and 70 per cent of the grate surface. Air should also be admitted above the fire, especially when each charge of fresh coal is fired. The air is usually admitted through holes in the door or in the furnace front. As the air is more efficient when heated, it can be made to circulate through a space left in the brick setting or be drawn from the ash-pit. In such cases it can be admitted through holes in the furnace sides or through a hollow bridge wall. The bridge of brick can be made hollow and draw its supply of air from the ash-pit, always under control by a damper, whose handle reaches back to the fire-room front. Some of the brick courses on the combustion-chamber side and near the top can be set about $\frac{1}{4}$ -inch apart without cement between them. The heated air can thus pass out of these openings in fine streams and mix with the products as they pass over the wall.

Probably the best arrangement is to admit the air both through the door openings and at the bridge wall, thus facilitating the mixing by adopting a number of openings. A steam jet may be utilized to insure a thorough mixing, but such an arrangement is not economical.

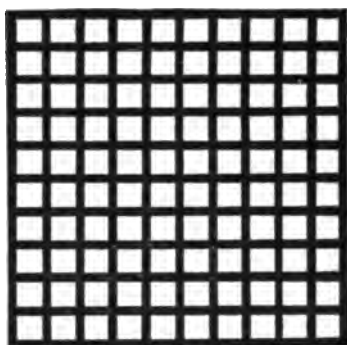
When coals are very smoky they should be fired in small quantities at frequent intervals on the "alternate" firing plan. In cases of lack of grate area, necessitating high rates of combustion, little can be done when the design is defective, except a general remodelling of the furnace.

The degree of smoke produced at any instant can be easily

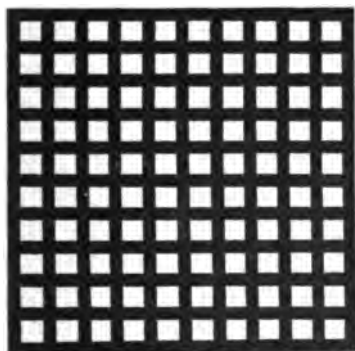
and well recorded by using Prof. Ringelmann's smoke-scales, described in *Engineering News*, 11 November, 1897. The cards should be about 8 inches square, and can be reproduced by a draftsman according to the following scheme, Fig. 152:*



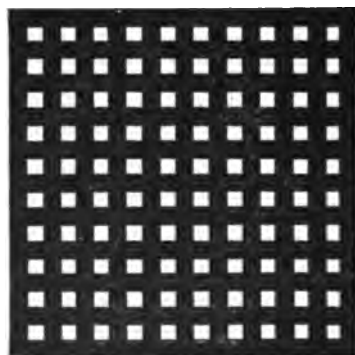
No. 1.



No. 2.



No. 3.



No. 4.

FIG. 152.—Prof. Ringelmann's Smoke-scales.

Card No. 0: All white.

Card No. 1. Black lines, 1 mm. thick, 10 mm. apart, leaving spaces 9 mm. square.

Card No. 2. Black lines, 2.3 mm. thick, leaving spaces 7.7 mm. square.

Card No. 3. Black lines, 3.7 mm. thick, leaving spaces 6.3 mm. square.

* Bryan Donkin advocated the use of tinted cards, each having a flat wash of gray corresponding to the effect of the lines on the Ringelmann cards.

Card No. 4. Black lines, 5.5 mm. thick, leaving spaces 4.5 mm. square.

Card No. 5. All black.

The observer glances from the smoke issuing from the stack to the cards, and determines which card most nearly corresponds with the color of the smoke, and makes a record accordingly, noting the time when the observation was made. Observations should be made continuously during, say, one minute and the estimated average density during that minute recorded, and so on, records being made once every minute. The average of all the records taken is the average for the smoke density. When these minute records are taken over a sufficiently long period, the whole can be plotted on section paper to show by a curve or broken line how the smoke density varied during that period.

The following is an extract from a report* to the State Board of Health of Tennessee on "Smoke Prevention," by Prof. Olin A. Landreth, of Vanderbilt University, 1893:

"When fresh coal is thrown on a bed of incandescent coal, or is otherwise highly heated, there immediately begins the distillation of the more volatile portions of the hydrocarbons in the coal, which distilled matter is burned if the temperature is high enough and a sufficient supply of oxygen is present, but which passes up the chimney as yellowish fumes if either of these two essential conditions of combustion is wanting. As the fresh coal becomes more highly heated the less volatile hydrocarbons are distilled, and these being, chemically speaking, unstable compounds, are decomposed or disassociated by the heat at a temperature much below that at which the carbon thus liberated combines with oxygen in combustion. The temperature necessary for combustion of this free carbon is very high, approximately 2000 degrees Fahr., and hence there is a wide margin of opportunity for this portion of the carbon to escape unburned, as this temperature is somewhat difficult to maintain in the mass of gaseous matter above the coal.

"It is this free, unburned carbon in a finely divided state which produces the bright, luminous flame, and which, when cooled, produces the black clouds of smoke that issue from the chimney and which afterward settle as soot. After the volatile matter is all driven off, there still remains the fixed carbon, which now is in

* Engineering News, 8 June, 1893.

the form of coke. This gives but little flame, and no smoke in burning, as the particles are not detached from the solid mass till combustion takes place.

"The causes of smoke may, from the foregoing description, be stated to be either

"(1) An insufficient amount of oxygen for the perfect combustion of these combustibles; or

"(2) An imperfect mixture or distribution of the oxygen with the combustibles, even though present in sufficient quantity; or

"(3) A temperature too low to ignite the distilled volatile matter and the separated free carbon when properly mixed with the air.

"In the ordinary boiler furnace, as generally constructed and fired, the conditions are very unfavorable for perfect combustion during the period in which the volatile matter is driven off from each charge of coal. When the fixed carbon stage is reached there is but little difficulty in maintaining perfect combustion, but when a fresh charge of coal is added the difficulties reappear; the air-supply, if not in excess during the burning of the previous incandescent coal, will now be in deficit, since the distillation of the volatile matter calls for an increased amount of air, while, in fact, the greater depth of coal now on the grate actually reduces the supply.

"If an additional supply is admitted through the furnace doors, it will be cold, and cannot be thoroughly mixed with the combustible gases. So with the temperature; if high enough before charging, it is now much lower owing to the cooling effects of the cold air rushing in when the doors are opened, of the mass of cold coal, of the evaporation of the moisture in the coal and to the distillation of the volatile matter, so that by the time a high temperature is needed to burn the free carbon the furnace is at its coldest.

"In fulfilling the requirements of sufficiency of supply, and thoroughness of mixing the air with the combustible gases, it must be noted that the conditions should not be secured by a reckless surplus of air, as this carries away useful heat which is not only a loss in itself, but may, and often does, result in lowering the temperature of the combustible gases below their temperature of ignition, thus causing the escape of unburned fuel. Owing to the difficulty of effecting such a thorough mixture, so as to bring to

each combustible particle just its proper amount of air, it is necessary to provide a surplus of air, but this should be considered as an evil to be kept at a minimum by the most thorough mixing possible.

"Passing to the means of accomplishing combustion without smoke production, it is safe to say that, so far as it pertains to steam-boilers, the object must be attained by one or more of the following agencies:

"1. By the proper design and setting of the boiler plant. This implies proper grate area, sufficient draft, the necessary air admission space between grate-bars and through furnace, and ample combustion room under boilers.

"2. By that system of firing that is best adapted to each particular furnace to secure the perfect combustion of bituminous coal. This may be either (a) 'Coke firing,' or charging all coal into the front of the furnace until partially coked, and then pushing back or spreading; or (b) 'Alternate side-firing'; or (c) 'Spreading,' by which the coal is spread over the whole grate area in thin, uniform layers at each charging.

"3. The admission of air through the furnace door, bridge wall or side walls.

"4. Steam jets and other artificial means of thoroughly mixing the air and combustible gases.

"5. Prevention of the cooling of the furnace and boilers by the inrush of cold air when the furnace doors are opened for charging coal and handling the fire.

"6. Establishing a gradation of the several steps of combustion, so that the coal may be charged, dried and warmed at the coolest part of the furnace, and then moved by successive steps to the hottest place, where the final combustion of the coked coal is completed, and compelling the distilled combustible gases to pass through the hottest part of the fire.

"7. Preventing the cooling by radiation of the unburned combustible gases until perfect mixing and combustion have been accomplished.

"8. Varying the supply of air to suit the periodic variation in demand.

"9. The substitution of a continuous uniform feeding of coal instead of intermittent charges.

"10. Down-draft burning, or causing the air to enter above the grates and pass down through the coal, carrying the distilled products down to the high-temperature plane at the bottom of the fire.

"The number of smoke-prevention devices are legion. The scope of the present paper renders anything more than a brief classification of their principles of working impossible. These are:

"(a) Mechanical stokers, which automatically deliver the fuel in a crushed or finely divided state into the furnace at a uniform rate, and also keep the fire clean by a slow but constant motion of the individual sections of the grate. They accomplish their object by means of agencies 5, 6 and 9 of the foregoing list. They affect a very material saving in the labor of firing, and are efficient smoke preventers when not pushed above their capacity, and when the coal does not cake badly. They are rarely susceptible to the sudden changes in the rate of firing frequently demanded in service.

"(b) Air-flues in side walls, bridge wall and grate-bars, through which air, when passing, is heated (agency 3). The results are always beneficial, but the flues are difficult to keep clean and in order.

"(c) Coking arches, or spaces in front of the furnace arched over, in which the fresh coal is coked, both to prevent cooling of the distilled gases and to force them to pass through the hottest part of the furnace, just beyond the arch (agencies 6 and 7). The results are good for normal conditions, but ineffective when the fires are forced. The arches also are easily burned out and injured by working the fire.

"(d) Dead-plates, or a portion of the grate next to the furnace doors reserved for warming and coking the coal before it is spread over the grate (agency 6). These give good results when the furnace is not forced above its normal capacity. This embodies the method of 'coke-firing' mentioned above.

"(e) Down-draft furnaces, or furnaces in which the air is supplied to the coal above the grate, and the products of combustion are carried away beneath the grate, thus causing a downward draft through the coal, carrying the distilled gases down the highly heated incandescent coal at the bottom of the layer of coal on the grate (agency 10). In this furnace the grate-bars must be kept cool by the cir-

culatation of water through them, as they have to bear the hottest portion of the flame.

"(f) Steam jets to draw air in, or inject air into the furnace above the grate, and also to mix the air and combustible gases together (agency 4). A very efficient smoke preventer, but one liable to be wasteful of fuel by inducing too rapid a draft.

"(g) Baffle plates placed in the furnace above the fire, to aid in mixing the combustible gases with the air (agency 4).

"(h) Double furnaces, of which there are two entirely different styles, the first of which places the second grate below the first grate; the coal is coked on the first grate; during the process the distilled gases are made to pass over the second grate, where they are ignited and burned; the coke from the first grate is dropped on to the second grate (agencies 6 and 7). A very efficient and economical smoke preventer, but rather complicated to construct and maintain. In the second form, the products of combustion from the first furnace pass through the grate and fire of the second, each furnace being charged with fresh fuel when needed, the latter generally with a smokeless coal or coke. An irrational and unpromising method.

"It is no longer a problem whether smoke can be prevented or not. This has been settled conclusively in the affirmative in a number of localities where proper laws for the abatement of smoke have been passed and enforced."

CHAPTER XVI

TESTING. BOILER COVERINGS. CARE OF BOILERS

Object of Testing New Boilers. Hydraulic Pressure. Methods Adopted. Measuring for Changes of Form. Limit of Test Pressure. Testing for Steam Leaks. Boiler Trials. Directions for Calculating Some Results. Boiler and Pipe Coverings. Heat Losses. Savings. Care of Boilers.

Testing. Boilers should always be tested before being accepted from the maker, but it should be remembered that the test is for the purpose of exposing faults, defects or leaks rather than of proving the strength of the structure. Many a good boiler has been ruined by being overstrained during its initial test.

Testing is always done by the application of pressure. As testing by steam is dangerous and not to be tolerated, hydraulic pressure has been almost universally adopted. The boiler can be filled with water, the valve closed, and a light fire built so as to warm the water, the pressure due to the expansion being noted on the gauge. When the required pressure has been reached, the valve should be slightly opened so as to maintain it or relieve it. As by this method it is difficult to control the pressure and make furnace measurements, a simpler plan is to fill the boiler with hot water and maintain the pressure for a few minutes by means of a pump. If the pressure falls rapidly there is indication of a leak, for which search should be made.

While under pressure, the boiler should be very closely examined for change of form. Careful measurements should be made before, during and after the test, and any change of form noted. If any permanent set is detected, care must be used to determine if it is due to an excess of the elastic limit of the material or to a tightening of the joints of stays and braces. If the flues show any tendency to flatten, such results must always be treated with great caution, since the defect is liable to become aggravated. Too

hot water cannot be used if accurate measurements are to be made, unless the boiler be heated before the first measurements are taken.

The limiting pressure for tests is usually placed at one and one-half the highest steam pressure to be carried. Many condemn this measure of test pressure and advocate some fixed increase above the highest working pressure as being more equitable; as for instance a test pressure to be the highest working pressure plus 90 pounds on the square inch. However, it is best not to allow, even in the best made boilers, a test pressure to exceed 40 per cent of the calculated strength of the weakest riveted joint.

Before a boiler is finally covered with lagging, but after it has been set and all connected, steam should be raised and leaks searched for. Most new boilers leak under steam, even after having proved tight under an hydrostatic test. Generally small leaks will close of their own accord, although ordinary leaks require calking after the steam pressure has been relieved. If the leaks are due to a defect in design, their closure is frequently a difficult matter. Care in working out details and in construction will be repaid many times over in preventing such annoyances.

Boiler Trials. Boilers are frequently tested when completed for obtaining data as to their actual performance. Such tests are termed "trials." The data should be collected by trained assistants and only calibrated instruments used. The object for which the trial is being made should always be clearly defined in advance, and all data bearing on the desired result be recorded by the assistants, who should work according to some prearranged plan.

For full information of how to carry out a boiler trial, reference is made to the "Report of the Committee on the Revision of the Society Code of 1885, relative to a Standard Method of Conducting Steam-boiler Trials," being the code of the American Society of Mechanical Engineers and published in the Society's Transactions, Vol. XXI, 1900. As this report is too voluminous to reprint in full, and as copies are obtainable from the Secretary of the Society, the following may prove useful, being an extract from "Experimental Mechanics," by D. S. Jacobus, Stevens Institute Indicator, and from the Am. Soc. M. E. Code.

Directions for Calculating Some Results Derived from Trial.
Total combustible = total dry coal consumed minus weight of refuse.

(Ordinarily the refuse is the ash and unburned coal raked out from the ash-pit.)

Per cent of ash = weight of ash $\times 100 \div$ total coal consumed.

Rate of combustion in pounds per sq. ft. of grate per hour = total coal consumed in lbs. divided by the length of test in hours and the grate area in square feet.

The weight of water evaporated at actual boiler pressure is the total amount of water fed to the boilers less the entrained water. The total weight of entrained water cannot be determined with a calorimeter having the usual form of collecting nipple arranged so as to draw out a small amount of steam from the steam-main, because the sample of steam collected and passed into the calorimeter may not be an average of the total amount flowing through the steam-main. Small throttling calorimeters are reliable, however, in showing whether the steam contains a considerable amount of moisture or is practically dry, and this is especially so if they are used in combination with an adjustable nozzle, which can be made to draw out a sample from any desired point in the cross-section of the pipe. To allow properly for the entrained water, the entire amount should be separated from the steam and weighed, or the entire mass of steam may, in special cases, be passed through a throttle-valve and exhausted at atmospheric pressure, and from the temperature of steam after throttling, the percentage of moisture can be calculated in the same way as for a throttling calorimeter. It must not be inferred from this that small throttling calorimeters are not useful in boiler tests. They should always be applied, if the steam is not found to be superheated; and if they indicate dry steam for samples taken from all sections of the pipe, or when a fixed nozzle is placed in such a position that any moisture in the steam would be thoroughly mingled and would be drawn into the nozzle, they prove definitely that the steam is dry.

Let t = temperature of feed-water,

H = the total heat of steam at the boiler pressure above 32° F.,

and W = weight of water actually evaporated;
then

$$\text{Equivalent evaporation from and at 212° F.} = W \frac{H - (t - 32)}{965.7},$$

where 965.7 is the latent heat of steam at atmospheric pressure. This equation is for dry saturated steam. If the steam is super-

heated the actual evaporation must be multiplied by the factor $\frac{H - (t - 32) + 0.48t'}{965.7}$, in which t' is the superheating in degrees Fahr.

The horse-power of the boiler, according to the standard of the American Society of Mechanical Engineers, is found by dividing the equivalent evaporation from and at 212° F. by 34.5.

The following is the formula for calculating the percentage of moisture in the steam when a throttling calorimeter is used:

$$w = 100 \times \frac{H - h - k(T - t)}{L},$$

in which w = percentage of moisture in the steam, H = total heat, L = latent heat per pound of steam at the pressure in the steam-pipe, h = total heat per pound of steam at the pressure in the discharge side of the calorimeter, k = specific heat of superheated steam, T = temperature of the throttled and superheated steam in the calorimeter, and t = temperature due to the pressure in the discharge side of the calorimeter, = 212° F. at atmospheric pressure. Taking $k = 0.48$ and $t = 212$, the formula reduces to

$$w = 100 \times \frac{H - 1146.6 - 0.48(T - 212)}{L}.$$

A correction should be made for radiation from the surface of the instrument. This loss, according to George H. Barrus, amounts to about three-tenths of one per cent of moisture.

The following is the formula for calculating the moisture in the steam when a barrel calorimeter is used:

Let W = the original weight of the water in calorimeter,

w = the weight of water added to the calorimeter by blowing steam into the water,

t = total heat of water corresponding to initial temperature of water in calorimeter,

t_1 = total heat of water corresponding to final temperature in calorimeter,

T_1 = total heat in the water at the temperature due to the steam pressure,

(This is nearly equal to the temperature of the steam less 32 degrees.)

H = total heat of steam due to the steam pressure,

x = pounds of steam blown into calorimeter,

y = percentage of priming,

k = degrees of superheating;

then

$$x = \frac{W(t_1 - t) - w(T_1 - t_1)}{(H - t_1) - (T_1 - t_1)},$$

$$y = 100 \cdot \frac{w - x}{w},$$

$$k = \frac{\frac{W(t_1 - t)}{w} + T_1 - H}{0.48}.$$

An approximate "heat balance," that is, a statement of the distribution of the heating value of the coal, may be reported in the form given on page 350.

The weight of air per pound of carbon burned is given by the formula

$$\frac{7N}{3(\text{CO}_2 + \text{CO})} \div 0.77;$$

where N , CO_2 , and CO represent the average per cents by volume of the several gases given by the analysis. This formula is approximate on account of the fact that the percentage of nitrogen in the coal is neglected. The error due to this cause is, however, a very small one, and the formula is much more accurate than a similar formula based on the ratio of the oxygen in the flue gas to the total carbon.

To find the amount of carbon burned per pound of coal, deduct from the total per cent of carbon determined by the analysis the percentage of unconsumed carbon contained in the ash. This percentage of unconsumed carbon in the ash is equal to the percentage of the ash as determined in the boiler test less the percentage of ash determined by analysis. For example, if the ash in the boiler test were 16 per cent and by analysis 12 per cent, the percentage of carbon unconsumed would be 4 per cent of the coal burned. If there were 80 per cent of carbon in the coal the

HEAT BALANCE, OR DISTRIBUTION OF THE HEATING VALUE OF THE COMBUSTIBLE.

Total Heat Value of 1 lb. of Combustible.....B. T. U.

	B. T. U.	Per Cent.
1. Heat absorbed by the boiler=evaporation from and at 212 degrees per pound of combustible $\times 965.7$.		
2. Loss due to moisture in coal=per cent of moisture referred to combustible $+100\times[(212-t)+966+0.48(T-212)]$ (t =temperature of air in the boiler-room, T =that of the flue gases).		
3. Loss due to moisture formed by the burning of hydrogen=per cent of hydrogen to combustible $+100\times 9\times[(212-t)+966+0.48(T-212)]$.		
4.* Loss due to heat carried away in the dry chimney gases=weight of gas per pound of combustible $\times 0.24\times(T-t)$.		
5.† Loss due to incomplete combustion of carbon= $\frac{\text{CO}}{\text{CO}_2+\text{CO}}\times\frac{\text{per cent C in combustible}}{100}\times 10,150$.		
6. Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemized if data are obtained from which they may be calculated.)		
Totals.....		100.00

* The weight of gas per pound of carbon burned may be calculated from the gas analyses as follows:

$$\text{Dry gas per pound carbon} = \frac{11\text{CO}_2 + 8\text{O} + 7(\text{CO} + \text{N})}{3(\text{CO}_2 + \text{CO})}, \text{ in which CO}_2, \text{CO, O and N are}$$

the percentages by volume of the several gases. As the sampling and analyses of the gases in the present state of the art are liable to considerable errors, the result of this calculation is usually only an approximate one. The heat balance itself is also only approximate for this reason, as well as for the fact that it is not possible to determine accurately the percentage of unburned hydrogen or hydrocarbons in the flue gases.

The weight of dry gas per pound of combustible is found by multiplying the dry gas per pound of carbon by the percentage of carbon in the combustible, and dividing by 100.

† CO_2 and CO are respectively the percentage by volume of carbonic acid and carbonic oxide in the flue gases. The quantity 10,150=Number heat-units generated by burning to carbonic acid one pound of carbon contained in carbonic oxide.

per cent of carbon burned per pound of coal and made to pass up the chimney would be $80-4=76$ per cent. To determine the pounds of air per pound of coal, multiply the pounds of air per pound of carbon by the amount of carbon burned per pound of coal, which, in the case just cited, would be 0.76.

Boiler and Pipe Coverings. All external surfaces of boilers, pipes, fittings, etc., from which loss of heat may occur should be well covered or insulated. The saving of the heat often pays for the covering within one year. Good coverings cost, in

place, from 17 cents to 25 cents per square foot of metal surface lagged.

The selection of a covering should depend upon absolute incombustibility and freedom from all substances which might cause corrosion. Coverings which carbonize after being in contact with a hot surface or which char when held in a flame are not fire-proof and, as a class, cannot be recommended. The best of the cork coverings, however, have proved satisfactory. Also, coverings which lose their shape and form after being some time in use, such as hair felt, cannot be classed as satisfactory or economical.

The conclusions reached by the Mutual Boiler Insurance Company, from tests made, were that "there were a sufficient number of safe, suitable and incombustible coverings for steam pipes and boilers . . . without giving regard to any of the composite coverings which contain combustible material in greater or lesser quantity, according to the integrity of the makers, and without giving regard to coverings which contain substances like the sulphate of lime, which may cause the dangerous corrosion of the metal against which it is placed." (Circular No. 6, Boston, 1898.)

The sectional coverings are the most convenient and can be removed with little injury. On large surfaces the sectional blocks are held in place by a wire netting, which should be woven from galvanized wire, and the blocks coated with a hard-finish plaster applied about one-quarter inch in thickness. This plaster coating can be painted. On pipes the sectional coverings can be wrapped with canvas having the edges sewed together, and the whole banded if so desired with brass or iron bands. These bands should be about 12 inches to 24 inches apart, according to size of pipe. Irregular pieces, fittings, valves, etc., are best insulated by a plastic coating applied thick enough to be even with the sectional pieces on the straight pipes adjacent.

A good covering does not need a layer of asbestos paper between it and the heated surface.

Coverings on boilers are best placed directly against the shell without an air space, so that any leak in a joint or rivet will reveal the spot and not trickle down the air space and appear at some distant point.

The following information has been abstracted from Transac-

tions American Society of Mechanical Engineers, Vol. XIX, 1898
("Protection of Steam-heated Surfaces," by C. L. Norton).

TABLE XXI

Specimen.	Name.	B. T. U. Loss per Sq. Ft. Pipe Sur- face per Min.	Ratio of Loss to Loss from Bare Pipe.	Thick- ness in Inches.	Weight in Ounces per Ft. of Length 4 In. Diam.
A.	Nonpareil Cork Standard ...	2.20	15.9	1.00	27
B.	" " Octagonal ...	2.38	17.2	.80	16
C.	Manville High Pressure.	2.38	17.2	1.25	54
D.	Magnesia.	2.45	17.7	1.12	35
E.	Imperial Asbestos.	2.49	18.0	1.12	45
F.	W. B.	2.62	18.9	1.12	59
G.	Asbestos Air Cell.	2.77	20.0	1.12	35
H.	Manville Infusorial Earth ...	2.80	20.2	1.50	
I.	" Low Pressure.	2.87	20.7	1.25	
J.	" Magnesia Asbestos.	2.88	20.8	1.50	65
K.	Magnabestos.	2.91	21.0	1.12	48
L.	Moulded Sectional.	3.00	21.7	1.12	41
O.	Asbestos Fire Board.	3.33	24.1	1.12	35
P.	Calcite.	3.61	26.1	1.12	66
	Bare Pipe.	13.84	100.0		

Specimen A consists of granulated cork pressed in a mould at high temperature and then submitted to a fire-proofing process. Made by Nonpareil Cork Co.

Specimen B is similar in composition, but is made up of several strips of cork instead of two semi-cylindrical sections. Made by Nonpareil Cork Co.

Specimen C is a sectional cover composed of an inner jacket of earthy material and an outer jacket of wool felt, the whole being $1\frac{1}{4}$ inches thick. Made by Manville Co.

Specimen D is a moulded sectional cover composed of about 90 per cent carbonate of magnesia. Made by Keasbey & Mattison Co.

Specimen E is essentially an air-cell cover, being composed of sheets of asbestos paper which has been indented before being laid up, the indentations serving to keep the thin sheets of paper from coming in close contact with one another, thereby causing a considerable amount of air to be held throughout the body of the cover. Made by H. F. Watson Co.

Specimen F is composed of a wool felt with a lining of asbestos paper. Made by H. F. Watson Co.

Specimen G is a cover made up of thin sheets of asbestos paper fluted or corrugated and stuck together with silicate of soda. Asbestos air-cell cover of the Asbestos Paper Co.

Specimen H is a plastic covering of infusorial earth, made by the Manville Co.

Specimen I is similar to specimen F, and made by the Manville Co.

Specimen J is a plastic cover made by the Manville Co.

Specimen K is a moulded cover containing about 45 per cent of carbonate of magnesia and a considerable percentage of carbonate of calcium, and made by the Keasbey & Mattison Co.

Specimen L is a moulded, sectional cover composed mainly of sulphate of calcium and some 25 per cent of carbonate of magnesia and has upon its outer surface a thick sheet of felt board.

Specimen O is similar to specimen G, except that it has larger cells and contains much more silicate of soda. It is very hard and strong. Made by Asbestos Paper Co.

Specimen P is a sectional moulded cover composed mainly of sulphate of calcium. It has an outer layer of felt board. Made by Philip Cary Co.

Purchasers should satisfy themselves that they are not buying under the name of "magnesia" a covering containing large quantities of sulphate of lime, which is liable to cause corrosion. Magnesia is a most effective non-conductor. Asbestos is merely an incombustible material in which air may be entrapped, but when not porous is a good conductor.

Table XXII gives the saving, in dollars, due to the use of the various covers.

Table XXIII shows that at the end of ten years the best of the covers tested will have saved \$46 more than the poorest. The difference between the several covers of the better grade is exceedingly small.

The following assumptions have been made in computing the tables:

That all the covers cost \$25 per hundred square feet applied. In case the saving due to a cover which costs \$20 instead of \$25 is desired, the simple addition to the final saving of the \$5 difference makes the necessary correction.

The money saving is computed on the following assumptions:

Coal at \$4 a ton evaporates 10 pounds of water per pound of coal. The pipes are kept hot ten hours a day three hundred and ten days a year. If computations are made, as is sometimes done, on an assumption that the pipes are hot twenty-four hours a day three hundred and sixty-five days a year, the saving is nearly three times that shown in Table XXII.

TABLE XXII

Specimen.	Name.	Loss B. T. U. 200 Lbs.	Saving B. T. U.	Saving per Year per 100 Sq. Ft.
A.	Nonpareil Cork Standard. . .	2.20	11.64	\$37.80
B.	Nonpareil Cork Octagonal. . .	2.38	11.46	37.20
C.	Manville Sectional, H. P. . . .	2.38	11.46	37.20
D.	Magnesia.	2.45	11.39	36.90
E.	Imperial Asbestos.	2.49	11.35	36.80
F.	W. B.	2.62	11.22	36.40
G.	Asbestos Air Cell.	2.77	11.07	36.00
H.	Manville Infusorial Earth. . . .	2.80	11.04	35.85
I.	Manville Low Pressure.	2.87	10.97	35.65
J.	Manville Magnesia Asbestos. . .	2.88	10.96	35.60
K.	Magnabestos.	2.91	10.93	35.50
L.	Moulded Sectional.	3.00	10.84	35.20
O.	Asbestos Fire Board.	3.33	10.51	34.20
P.	Calcite.	3.61	10.23	33.24
	Bare Pipe.	13.84	0.00	

Inspection of Table XXIV shows the saving due to the use of hair felt outside of a standard magnesia cover.

In five years 100 square feet of hair felt saves \$7 more than its cost, and in ten years it saves \$20 above its cost.

The further saving due to a second inch outside the first is \$8 in ten years. Of course the well-known tendency of hair felt to deteriorate should be considered.

In the case of Nonpareil Cork, increasing the thickness from 1 to 2 inches raises the cost from about \$25 to \$35 per 100 square feet and increases the net saving in five years by \$10 and by \$30 in ten years. In other words, the second inch of material in use about pays for itself in two years, while the first pays for itself in about one year. The third inch does not increase the saving even in ten years. The second inch, therefore, more than pays for interest and depreciation, while the third fails to do this.

In the case of Asbestos Fire Board, a second inch in thickness causes a saving of \$20 in ten years, the third and fourth inches showing a loss.

In general it may be said, therefore, that if five years is the length of life of a cover, one inch is the most economical thickness, while a cover which has a life of ten years may to advantage be made 2 inches thick.

TABLE XXIII
NET SAVING PER 100 SQ. FT.

Specimen.	Name.	1 Year.	2 Years.	5 Years.	10 Years.
A.....	Nonpareil Cork Standard.....	\$12.80	\$50.60	\$164.00	\$353.00
B.....	Nonpareil Cork Octagonal.....	12.20	49.40	161.00	347.00
C.....	Manville Sectional High Pressure	12.20	49.40	161.00	347.00
D.....	Magnesia.....	11.90	48.80	159.50	344.00
E.....	Imperial Asbestos.....	11.80	48.60	159.00	343.00
F.....	W. B.....	11.40	47.80	157.00	339.00
G.....	Asbestos Air Cell.....	11.00	47.00	155.00	335.00
H.....	Manville Infusorial Earth.....	10.85	46.70	154.25	333.00
I.....	Manville Low Pressure.....	10.65	46.30	153.75	331.00
J.....	Manville Magnesia Asbestos.....	10.60	46.20	153.00	331.00
K.....	Magnabestos.....	10.50	46.00	152.50	330.00
L.....	Watson's Moulded Sectional.....	10.20	45.40	151.00	327.00
O.....	Asbestos Fire Board.....	9.20	43.40	146.00	317.00
P.....	Calcite.....	8.24	41.48	141.20	307.00
Q.....	Bare Pipe.....				

TABLE XXIV
VARIATIONS IN THICKNESS, ETC.

Specimen.	Saving in B. T. U. per Sq. Ft. per Minute.	Saving in Dollars per 100 Sq. Ft. per Year.	Net Saving.				Approximate Cost.
			1 Year.	2 Years.	5 Years.	10 Years.	
Magnesia, 1½ inches thick.....	11.62	\$37.75	\$7.75	\$45.50	\$159	\$347	\$30
Magnesia, 1½ inches thick and 1 inch of hair felt.....	12.38	40.22	5.22	45.44	166	367	35
Magnesia, 1½ inches thick and 2 inches of hair felt.....	12.77	41.50	1.50	43.00	167	375	40
Nonpareil cork:							
1 inch.....	11.64	37.80	12.80	50.60	164	353	25
2 inches.....	12.84	41.75	7.75	48.50	174	383	35
3 inches.....	12.94	42.05	7.95	34.10	160	370	50
Fire board:							
1 inch.....	10.54	34.20	9.20	43.40	146	317	25
2 inches.....	11.48	37.25	2.25	39.50	151	337	35
3 inches.....	11.70	38.00	12.00	26.00	140	330	50
4 inches.....	11.83	38.40	26.60	11.80	127	319	65

TABLE XXV
LOSS OF HEAT AT 200 POUNDS FROM BARE PIPE

Condition of Specimen.	B. T. U. Lost per Sq. Ft. per Minute.
New pipe.....	11.96
Fair condition.....	13.84
Rusty and black.....	14.20
Cleaned with caustic potash inside and out.....	13.85
Painted dull white.....	14.30
Painted glossy white.....	12.02
Cleaned with potash again.....	13.84
Coated with cylinder oil.....	13.90
Painted dull black.....	14.40
Painted glossy black.....	12.10

TABLE XXVI
VARIATION OF HEAT LOSS WITH PRESSURE

Pressure.	Bare Pipe. Loss B. T. U. per Sq. Ft. per Minute.
340	15.97
200	13.84
100	8.92
80	8.04
60	7.00
40	5.74

TABLE XXVII
MISCELLANEOUS SUBSTANCES

Specimen.	B. T. U. per Sq. Ft. per Minute at 200 Lbs.	Saving in One Year per 100 Sq. Ft. Pipe.
Box A.		
1 with sand.....	3.18	\$34.60
2 with cork, powdered.....	1.75	39.40
3 with cork and infusorial earth.....	1.90	38.90
4 with sawdust.....	2.15	37.90
5 with charcoal.....	2.00	38.50
6 with ashes.....	2.46	36.90
Brick wall 4 inches thick.....	5.18	28.80
Pine wood 1 inch ".....	3.56	33.80
Hair felt 1 " ".....	2.51	36.80
Cabot's seaweed quilt.....	2.78	35.90
Spruce 1 inch thick.....	3.40	33.90
" 2 inches ".....	2.31	37.50
" 3 " ".....	2.02	38.50
Oak 1 inch thick.....	3.65	33.10
Hard pine 1 inch thick.....	3.72	32.90

The box A referred to in the table is a $\frac{7}{8}$ -inch pine box, large enough to surround the pipe and leave a 1-inch minimum space at its four sides.

Table XXVIII is a comparison of results obtained by D. S. Jacobus and George H. Barrus from experiments made separately in 1901.

TABLE XXVIII

AVERAGE RESULTS OBTAINED FOR 2-INCH PIPES AS MEASURED BY THE PERCENTAGE OF THE HEAT RADIATED BY THE BARE PIPES THAT WAS SAVED BY APPLYING THE COVERINGS

Covering.	Tests by Barrus.	Tests by Jacobus.
Remanit for high-pressure steam.	86.9
Hair Felt.	86.0
Johns' Asbesto-Sponge-Hair Felt, 3-ply.	85.1
Johns' Asbesto-Sponge-Hair Felt, 2-ply.	84.4
Asbesto-Sponge Felted (Sectional).	84.2	84.9
Remanit for low-pressure steam.	84.4
Keasbey & Mattison's Magnesia.	83.4	83.2
Johns' Asbestos Fire Felt (Navy Brand).	81.1	82.3
Johns' Asbestocel.	76.3	77.2
New York Air Cell.	75.9
Carey's Moulded.	74.9
Johns' Moulded.	74.8
Gast's Ambler Air Cell.	74.4	74.4
Johns' Asbestos Fire Felt.	73.1

It may be well to explain that the Johns' Asbestos-Sponge-Hair Felt covering is made up of layers of fabric composed of fiberized asbestos and carded hair, felted and laminated.

Care of Boilers. The care of steam-boilers is all important. They are often sadly neglected, although they should receive as much and as careful attention as any part of the plant.

A boiler should be so designed and constructed that it can be inspected at all parts, and the owner should see that it is inspected by some competent person. A boiler which cannot be so inspected because of its arrangement or setting should be handled with caution.

All internal fittings, as fusible plugs, feed-pipes, water-alarms, sediment-collectors, and the like should be examined occasionally to see that they are not loose and are in good working order. In fact, the time and attention given to a steam-boiler will be repaid many fold through the increase in its life, its safety and its economy.

At periods of examination and cleaning a rigid search should be made for corrosion and grooving, both externally and internally. Corrosion may be expected anywhere, but points especially liable to attack are at the water-line, at the supports and at places touched by brickwork or ashes. Grooving can be looked for where expansion stresses cause a bending action, as at lap-seams and near stay-ends.

The following are the rules for management and care, as issued by the Hartford Steam-boiler Inspection and Insurance Company:

1. *Condition of Water.*—The first duty of an engineer, when he enters his boiler-room in the morning, is to ascertain how many gauges of water there are in his boilers. *Never unbank nor replenish the fires until this is done.* Accidents have occurred, and many boilers have been entirely ruined from neglect of this precaution.

2. *Low Water.*—In case of low water, immediately cover the fires with ashes, or, if no ashes are at hand, use *fresh coal*, and close ash-pit doors. Don't turn on the feed under any circumstances, nor tamper with or open the safety-valve. Let the steam outlets remain as they are.

3. *In Case of Foaming.*—Close throttle, and keep closed long enough to show true level of water. If that level is sufficiently high, feeding and blowing will usually suffice to correct the evil. In case of violent foaming, caused by dirty water, or change from salt to fresh or *vice versa*, in addition to the action above stated, check draft and cover fires with fresh coal.

4. *Leaks.*—When leaks are discovered they should be repaired as soon as possible.

5. *Blowing-off.*—Clean furnace and bridge wall of all coal and ashes. Allow brickwork to cool down for two hours at least before opening blow. A pressure exceeding 20 lbs. should not be allowed when boilers are blown out. Blow out at least once in two weeks. In case the feed becomes muddy, blow out six or eight inches every day. When surface blow-cocks are used, they should be often opened for a few moments at a time.

6. *Filling Up the Boiler.*—After blowing down *allow the boiler to become cool* before filling again. Cold water pumped into hot boilers is very injurious from sudden contraction.

7. *Exterior of Boiler.*—Care should be taken that no water

comes in contact with the exterior of the boiler, either from leaky joints or other causes.

8. *Removing Deposit and Sediment.*—In tubular boilers the hand holes should be often opened, all collections removed, and fire-plates carefully cleaned. Also, when boilers are fed in front and blown off through the same pipe, the collection of mud or sediment in the rear end should be often removed.

9. *Safety-valves.*—Raise the safety-valves cautiously and frequently, as they are liable to become fast in their seats and useless for the purpose intended.

10. *Safety-valve and pressure-gauge.*—Should the gauge at any time indicate the limit of pressure allowed by this company, see that the safety-valves are blowing off. In case of difference, notify the company's inspector.

11. *Gauge-cocks, Glass Gauge.*—Keep gauge-cocks clear and in constant use. Glass gauges should not be relied on altogether.

12. *Blisters.*—When a blister appears there must be no delay in having it carefully examined and *trimmed* or *patched*, as the case may require.

13. *Clean Sheets.*—Particular care should be taken to keep sheets and parts of boilers exposed to the fire perfectly clean, also all tubes, flues and connections well swept. This is particularly necessary where wood or soft coal is used for fuel.

14. *General Care of Boilers and Connections.*—Under all circumstances keep the gauges, cocks, etc., clean and in good order, and things generally in and about the engine and boiler-room in a neat condition.

15. *Getting Up Steam.*—In preparing to get up steam after boilers have been open or out of service, great care should be exercised in making the manhole and handhole joints. Safety-valve should then be opened and blocked open and the necessary supply of water run in or pumped into the boilers until it shows at second gauge in tubular and locomotive boilers; a higher level is advisable in vertical tubulars as a protection to the top ends of tubes. After this is done, fuel may be placed upon the grate, dampers opened and fires started. If chimney or stack is cold and does not draw properly, burn some oily waste or light kindlings at the base. Start fires in ample time so it will not be necessary to urge them unduly. When steam issues from the safety-valve, lower it

carefully to its seat and note pressure and behavior of steam-gauge.

If there are other boilers in operation and stop-valves are to be opened to place boilers in connection with others on a steam pipe line, watch those recently fired up until pressure is up to that of the other boilers to which they are to be connected; and, when that pressure is attained, open the stop-valves *very slowly and carefully*.

APPENDIX A

Superheated Steam.* When steam at any pressure contains the amount of heat necessary to maintain its condition as the vapor of water, it contains no moisture, and is said to be "saturated steam" or "dry saturated steam." When it contains moisture, it is said to be "wet steam." In the latter expression, the word

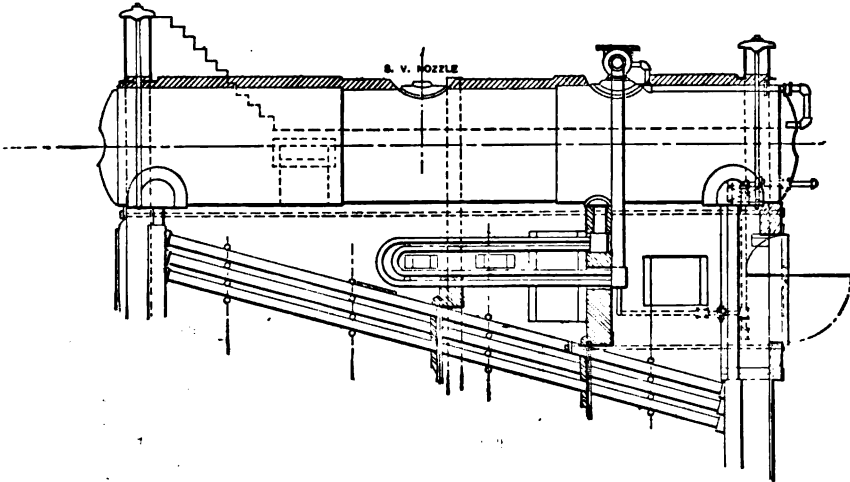


FIG. 153.—Superheater Attached to a Babcock and Wilcox Boiler.

steam means the apparent evaporation, since the wetness contained is not steam, but finely divided particles of water which lack the required latent heat.

When dry saturated steam has been heated to a temperature in excess of the boiling-point corresponding to its pressure, it is called "superheated steam."

The object of using superheated steam is threefold: It acts more nearly as a perfect gas; it is a poorer conductor of heat than

* See Trans. American Society of Mechanical Engineers.

wet steam; and it can lose heat through performance of work and radiation before it becomes saturated or wet. The cylinder con-

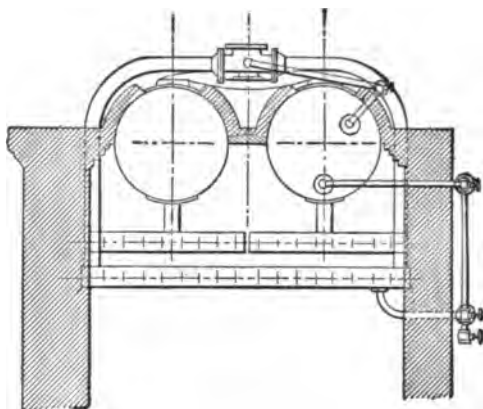


FIG. 153a.—Section of Fig. 153.

densation, which constitutes one of the chief losses of heat in a working engine, is, therefore, greatly reduced. In a turbine, the

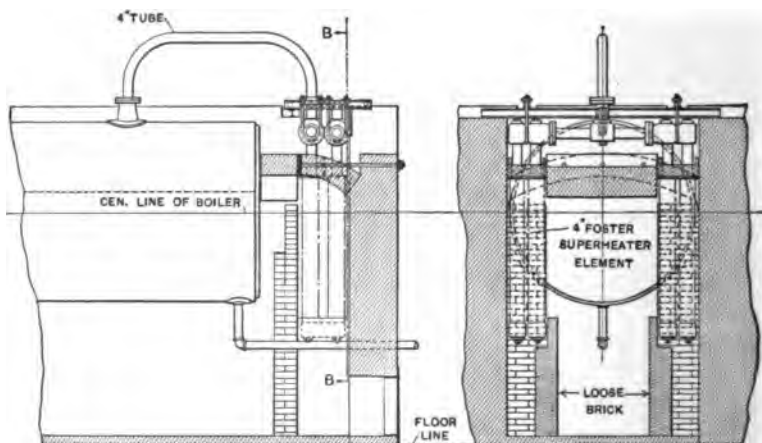


FIG. 154 —Foster's Superheater attached to a Fire-tube Boiler.

FIG. 154a.—Section BB.

reduction in steam consumption is about the same as in a reciprocating engine. It requires less heat to generate superheated steam than it does dry saturated steam at the same temperature. The

greatest economies are reached with superheated steam in simple-expansion engines and in engines of poor design, since one of the objects of multiple expansion, steam jackets, reheaters, and similar complications is to reduce the initial condensation.

Experience has shown that the comparative economy of superheated over saturated steam diminishes rapidly beyond the point of double expansion. There is less advantage in using superheated steam in triple- or quadruple-expansion engines unless some gains

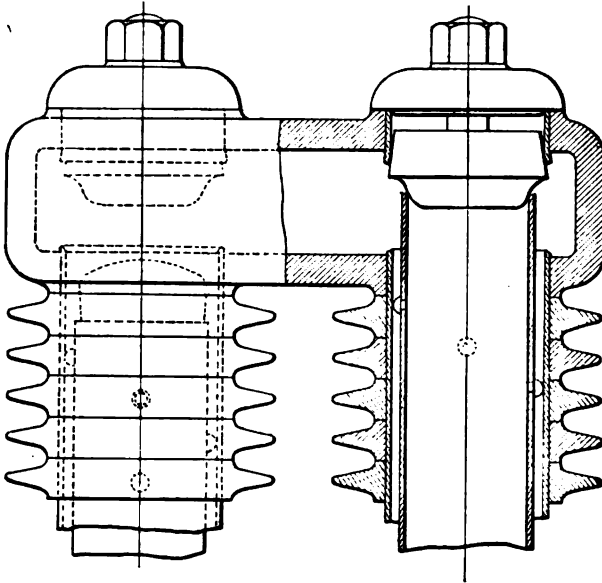


FIG. 155 —Detail of Return Header and Elements.

Portion of Foster Superheater showing ends of elements connected by return header. The elements consist of seamless drawn steel tubing protected by cast-iron rings shrunk on. Inner tubes are closed to steam, which is thus forced through thin annular spaces and rapidly superheated.

are sought other than a lessening of cylinder or initial condensation.

The economy derived by the proper use of superheated steam appears to vary according to conditions from nothing to 40 per centum of the fuel required. The maximum saving is found in slow-moving, simple engines, such as direct-acting pumps. It also appears when superheated steam is used that the weight of steam

consumed by an engine per unit of power becomes nearly uniform, and is therefore less dependent on the size of the engine.

The use of superheated steam is based on correct thermodynamic principles. The plant is complicated by the superheater, and the operating costs are increased by radiation losses, repairs, renewals to both engine and heater, interest and depreciation. The engine is subject to a change of form due to uneven expansion with high temperatures, and consequently

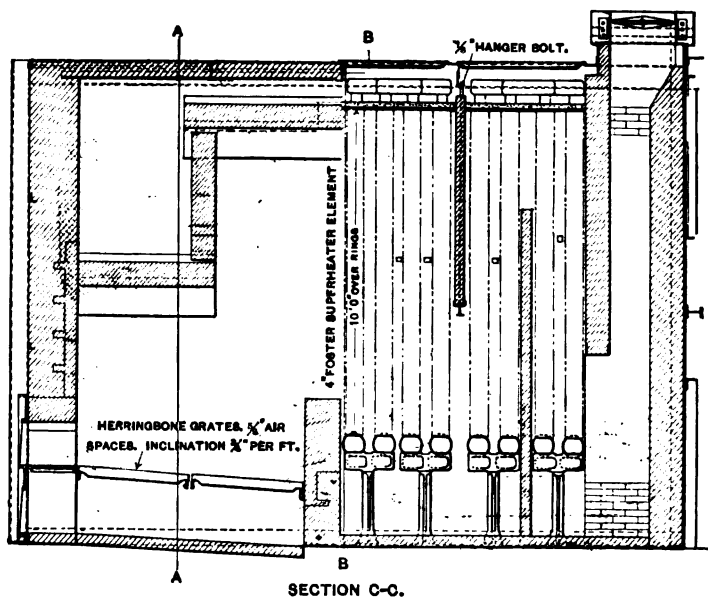


FIG. 156.—Foster's Superheater—Direct-fired type.

greater care must be exercised in the design of its parts. The effect of introducing superheat may simplify a plant by the elimination of other features. The specific heat of steam was determined by Regnault to be 0.48. Recent experiments show that it increases with the pressure and temperature; thus, at 150 pounds the specific heat is about 0.55. Superheated steam will lose heat as readily as it will receive it, which makes it essential that pipes and surfaces be well covered.

The greatest benefit with the least complication is obtained with steam heated to a temperature not in excess of 550° Fahr. at the

engine, and with pressures of about 160 pounds per square inch. The amount of superheating surface required is difficult to determine in advance. Practically it varies from about 10% to 100% of water-heating surface. The surface is made up of tubes of wrought-iron or steel. Wrought-iron and steel tubes seldom show pitting or corrosion, but must have a circulation through them or they will burn out. When not fitted with a cast-iron protecting

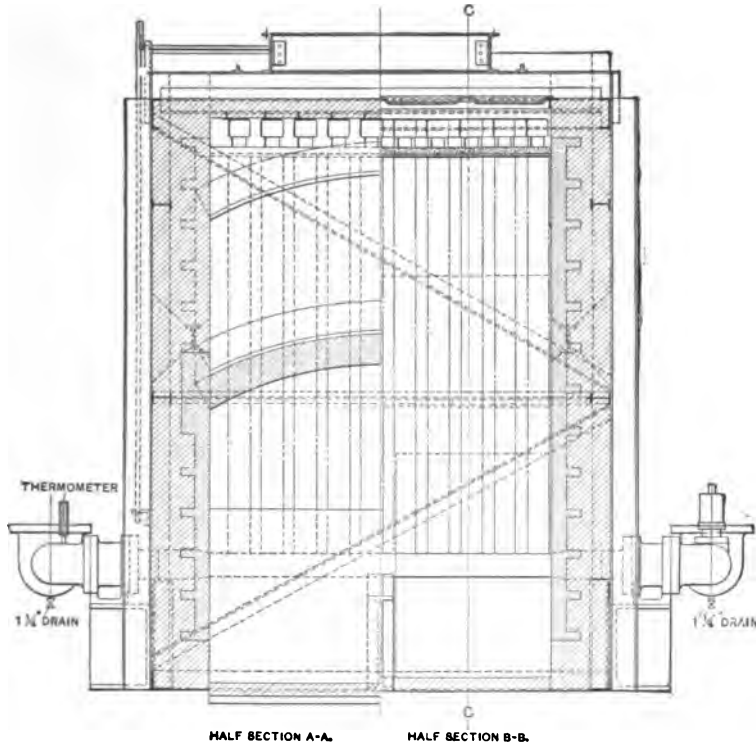


FIG. 156a.—Sections of Fig. 156.

cover, superheaters should be arranged to be flooded when not in use, and the water must be drained off before the circulation can be re-established. Screw joints should be avoided, but if they have to be used the threads should be cut as perfectly as possible and the joints made with a paste of plumbago mixed with linseed oil. Expanded ends are better than screw-joints when possible. The tubes are $1\frac{1}{2}$ inches, 2 inches, or 4 inches in diameter.

Superheaters fitted with cast-iron covers have an advantage due to the reserve of heat stored in the greater weight of metal, and the surface should be ribbed or fluted on the outside.

The cost of superheating surface is nearly the same as that for an equal amount of boiler-heating surface.

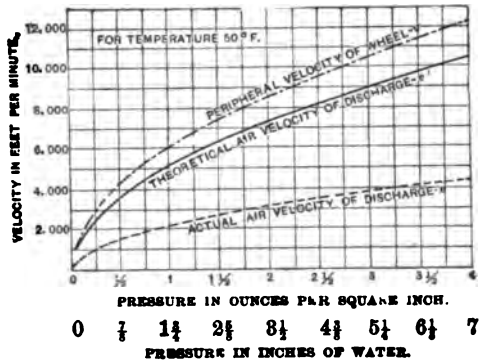
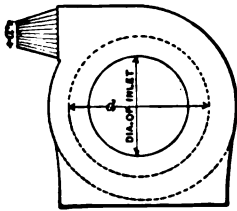
Figs. 153 and 153a illustrate a superheater attached to a water-tubular boiler of the Babcock & Wilcox design. Figs. 154 and 154a illustrate E. H. Foster's superheater attached to a fire-tube boiler.

Fig. 155 illustrates Foster's detail of return header protected by cast-iron rings. Figs. 156 and 156a illustrate Foster's direct-fired type of superheater.

APPENDIX B

FAN WHEEL WITH PERIPHERAL DISCHARGE

CALCULATIONS FOR CAPACITY, REVOLUTIONS, AND BRAKE HORSE-POWER



Q = Air discharged in cu. ft. per minute.

d = diameter of wheel in feet.

w = width of wheel at tip of blades in feet.

R = revolutions per minute.

V = peripheral velocity of wheel in feet per minute.

v' = theoretical velocity of air " " " "

v = actual velocity of air " " " "

D = density of air in pounds.

P = brake h. p. to drive wheel.

a = effective area of discharge or "blast area" in square feet.

Width of blades at tip is usually made about $0.4 \times d$.

Width of blades at widest part is made about $0.5 \times d$.

* When small volume of air, discharged at high pressure, is desired, the width is less.

* When large volume of air, discharged at low pressure, is desired, the width is greater.

$a = Q \div V$. By experiment, approximately $a = wd \div 3$. Therefore $a = 0.4d^2 \div 3$, from which d can be found.

$$v = 0.43 \times v', \quad V = 1.17v', \quad R = V \div \pi d.$$

Inlet area in sq. ft. $= 0.00054Q \div \sqrt{\text{water pressure in inches}}$, but should not exceed 40 per cent of disc area of side of wheel.

Outlet area in sq. ft. $= \text{constant} \times \text{inlet area}$.

For free discharge, constant varies from 1.0 to 1.25.

For restricted discharge, as into ducts, the fan should be calculated for a pressure equal to that at outlet plus friction.

Theoretical b. h. p. to drive fan wheel $= \frac{Q \times D \times v^2}{550 \times 2g}$; Q and v' being taken in feet per second and $Q = a \times V$.

$\therefore P = \text{theo. h. p.} \times 2$. The efficiency being taken at 50 per cent.

VALUES OF K FOR VARYING TEMPERATURES.

80° F. = 0.98	200° F. = 1.14	400° F. = 1.30
50° = 1.00	250° = 1.18	450° = 1.34
100° = 1.05	300° = 1.22	500° = 1.37
150° = 1.09	350° = 1.26	550° = 1.41

At any temperature F ., the velocities $= K \times \text{velocity at } 50^\circ \text{ F.}$

Therefore, work out fan problem with corrected velocities corresponding to the temperature.

* This is true when d or R is fixed, and V or discharge pressure is known. Usual Max. for $V = 6600$ ft. per min., but should not exceed 7200 ft. per min.

SATURATED-STEAM TABLES.

Absolute pressure, pounds per sq. inch.	Temperature of boiling-point, degrees F.	Heat of the liquid from 32° F.	Total heat from 32° F.	Latent heat.	Heat to overcome internal resistance.	Heat to overcome external resistance.	Thermal units actually contained in the steam above 32° F.	Weight of a cub. ft. in pounds.	Cub. ft. per pound.
<i>p</i>	<i>t</i>	<i>A</i>	<i>H</i>	<i>L</i>	<i>p</i>	<i>E</i>	<i>I = H - E</i>	<i>w</i>	<i>V</i>
1.0	102.0	70.0	1113.1	1043.0	981.1	61.9	1051.2	0.00229	334.6
2.0	126.3	94.4	1120.5	1026.1	961.9	64.2	1056.3	0.00576	173.6
3.0	141.6	109.8	1125.1	1015.3	949.5	65.8	1059.3	0.00844	118.4
4.0	153.1	121.4	1128.6	1007.2	940.4	66.8	1061.8	0.01107	90.31
5.0	162.3	130.7	1131.5	1000.8	933.1	67.7	1063.8	0.01366	73.22
6.0	170.1	138.6	1133.8	995.2	926.7	68.5	1065.3	0.01622	61.67
7.0	176.9	145.4	1135.9	990.5	921.4	69.1	1066.8	0.01874	53.37
8.0	182.9	151.5	1137.7	986.2	916.5	69.7	1068.0	0.02112	47.07
9.0	188.3	156.9	1139.4	982.5	912.4	70.1	1069.3	0.02374	42.13
10.0	193.2	161.9	1140.9	979.0	908.4	70.6	1070.3	0.02621	38.16
11.0	197.8	166.5	1142.3	975.8	904.8	71.0	1071.3	0.02866	34.88
12.0	202.0	170.7	1143.6	972.9	901.5	71.4	1072.2	0.03111	32.14
13.0	205.9	174.6	1144.7	970.1	898.4	71.7	1073.0	0.03355	29.82
14.0	209.6	178.3	1145.8	967.5	895.5	72.0	1073.8	0.03600	27.79
14.7	212.0	180.7	1146.6	965.8	893.5	72.3	1074.2	0.03758	26.64
15.0	213.0	181.8	1146.9	965.1	892.6	72.5	1074.4	0.03826	26.15
16.0	216.3	185.1	1147.9	962.8	890.0	72.8	1075.1	0.04067	24.59
17.0	219.4	188.3	1148.9	960.6	887.6	73.0	1075.9	0.04307	23.22
18.0	222.4	191.3	1149.8	958.5	885.3	73.2	1076.6	0.04547	22.00
19.0	225.2	194.1	1150.7	956.6	883.2	73.4	1077.3	0.04786	20.90
20.0	227.9	196.9	1151.5	954.6	881.0	73.6	1077.9	0.05023	19.91
21.0	230.5	199.5	1152.3	952.8	879.0	73.8	1078.5	0.05259	19.01
22.0	233.1	202.0	1153.0	951.0	877.0	74.0	1079.0	0.05495	18.20
23.0	235.5	204.5	1153.7	949.2	875.0	74.2	1079.5	0.05731	17.45
24.0	237.8	206.8	1154.4	947.6	873.2	74.4	1080.0	0.05966	16.78
25.0	240.0	209.1	1155.1	946.0	871.5	74.5	1080.6	0.06199	16.13
26.0	242.2	211.2	1155.8	944.6	869.9	74.7	1081.1	0.06432	15.55
27.0	244.3	213.4	1156.5	943.1	868.2	74.9	1081.6	0.06666	15.00

These Tables are reproduced from "Steam Engine Theory and Practice," by William Ripper, M.I.C.E., and are for the most part taken from Professor Peabody's valuable "Saturated Steam Tables," by kind permission of the author and publishers (Messrs. John Wiley and Sons, New York).

Absolute pressure, pounds per sq. inch.	Temperature of boiling-point, degrees F.	Heat of the liquid from 32° F.	Total heat from 32° F.	Latent heat.	Heat to overcome internal resistance.	Heat to overcome external resistance.	Thermal units actually contained in the steam above 32° F.	Weight of a cub. ft. in pounds.	Cub. ft. per pound.
<i>p</i>	<i>t</i>	<i>h</i>	<i>H</i>	<i>L</i>	<i>p</i>	<i>E</i>	$I = H - E$	<i>w</i>	<i>v</i>
28.0	246.4	215.4	1157.1	941.7	866.7	75.0	1082.1	0.06899	14.49
29.0	248.3	217.4	1157.7	940.3	865.1	75.2	1082.5	0.07130	14.03
30.0	250.3	219.4	1158.3	938.9	863.6	75.3	1083.0	0.07360	13.59
31.0	252.1	221.3	1158.8	937.5	862.0	75.5	1083.3	0.07590	13.18
32.0	254.0	223.1	1159.4	936.3	860.7	75.6	1083.8	0.07821	12.78
33.0	255.8	224.9	1159.9	935.0	859.2	75.8	1084.1	0.08051	12.41
34.0	257.5	226.7	1160.4	933.7	857.8	75.9	1084.5	0.08280	12.07
35.0	259.2	228.4	1161.0	932.6	856.6	76.0	1085.0	0.08508	11.75
40.0	267.1	236.4	1163.4	927.0	850.3	76.7	1086.7	0.09644	10.37
45.0	274.3	243.6	1165.6	922.0	844.8	77.2	1088.4	0.1077	9.287
50.0	280.8	250.2	1167.6	917.4	839.7	77.7	1089.9	0.1188	8.414
55.0	286.9	256.3	1169.4	913.1	834.9	78.2	1091.2	0.1299	7.696
60.0	292.5	261.9	1171.2	909.3	830.7	78.6	1092.6	0.1409	7.096
65.0	297.8	267.2	1172.7	905.5	826.5	79.0	1093.7	0.1519	6.583
70.0	302.7	272.2	1174.3	902.1	822.7	79.4	1094.9	0.1628	6.144
75.0	307.4	276.9	1175.7	898.8	819.1	79.7	1096.0	0.1736	5.762
80.0	311.8	281.4	1177.0	895.6	815.5	80.1	1096.9	0.1843	5.425
85.0	316.0	285.8	1178.3	892.5	812.1	80.4	1097.9	0.1951	5.125
90.0	320.0	290.0	1179.6	889.6	808.9	80.7	1098.9	0.2058	4.858
95.0	323.9	294.0	1180.7	886.7	805.8	80.9	1099.8	0.2165	4.619
100.0	327.6	297.9	1181.9	884.0	802.8	81.2	1100.7	0.2271	4.403
105.0	331.1	301.6	1182.9	881.3	799.9	81.4	1101.5	0.2378	4.206
110.0	334.6	305.2	1184.0	878.3	797.1	81.7	1102.3	0.2484	4.026
115.0	337.9	308.7	1185.0	876.3	794.4	81.9	1103.1	0.2589	3.862
120.0	341.0	312.0	1186.0	874.0	791.9	82.1	1103.9	0.2695	3.711
125.0	344.1	315.2	1186.9	871.7	789.4	82.3	1104.6	0.2800	3.572
130.0	347.1	318.4	1187.8	869.4	786.9	82.5	1105.3	0.2904	3.444
135.0	350.0	321.4	1188.7	867.3	784.7	82.6	1106.1	0.3009	3.323
140.0	352.8	324.4	1189.5	865.1	782.3	82.8	1106.7	0.3113	3.212
145.0	355.6	327.2	1190.4	863.2	780.2	83.0	1107.4	0.3218	3.107
150.0	358.3	330.0	1191.2	861.2	778.1	83.1	1108.1	0.3321	3.011
155.0	360.9	332.7	1192.0	859.3	776.0	83.3	1108.7	0.3426	2.919
160.0	363.4	335.4	1192.8	857.4	774.0	83.4	1109.4	0.3530	2.833
165.0	365.9	338.0	1193.6	855.6	772.0	83.6	1110.0	0.3635	2.751
170.0	368.3	340.5	1194.3	853.8	770.1	83.7	1110.6	0.3737	2.676
175.0	370.6	343.0	1195.0	852.0	768.2	83.8	1111.2	0.3841	2.608
180.0	373.0	345.4	1195.7	850.3	766.4	83.9	1111.8	0.3945	2.535
185.0	375.23	347.8	1196.4	848.6	764.6	84.0	1112.4	0.4049	2.470
190.0	377.4	350.1	1197.1	847.0	762.9	84.1	1113.0	0.4153	2.408
195.0	379.6	352.4	1197.7	845.3	761.1	84.2	1113.5	0.4257	2.349
200.0	381.7	354.6	1198.4	843.8	759.5	84.3	1114.1	0.4359	2.294
250.0	401.0	374.7	1204.2	829.5	744.5	85.0	1119.2	0.5393	1.854
300.0	417.4	391.9	1209.3	817.4	732.0	85.4	1123.9	0.6440	1.554
400.0	444.9	419.8	1217.7	797.9	712.3	86.2	1131.5	0.8572	1.167

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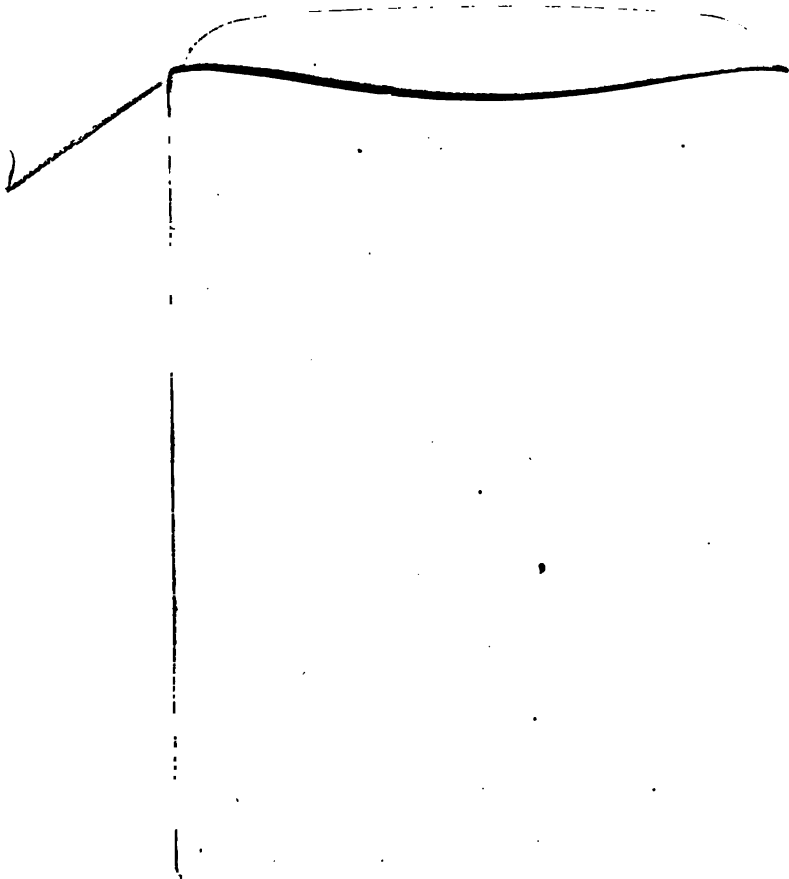




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